PUMPING BY COMPRESSED AIR

BY

EDMUND M. IVENS, B.E., M.E.

MEMBER OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

SECOND EDITION, REVISED

NEW YORK

JOHN WILEY & SONS, INC.

LONDON: CHAPMAN & HALL, LIMITED

1920

THE · PLIMPTON · PRESS
NORWOOD · MASS · U · S · A

PREFACE

Some years ago, when I first became interested in the subject of compressed-air pumping, I endeavored to obtain a book, or literature of some sort, that I could use as a guide in the design and installation of plants of this kind. I found comparatively little of any definite value; in fact the only information I could gather was obtained from air compressor manufacturers' catalogues, a few brief articles in technical society journals and engineering periodicals, and some data in works on compressed air which were either more or less a repetition of that contained in catalogues, or were of a purely theoretical nature.

It was evident that I would have to depend upon my own efforts and experiments in the field for any practical working data that I might need. I was very fortunate in that I had the opportunity to install and test a number of air lifts operating under a wide range of conditions and have consequently amassed a large volume of data. This data I thought of sufficient interest and value to condense and publish.

In preparing this book, I have endeavored to place in the hands of the student a comprehensive theoretical study of the subject, and at the same time instruct the operating engineer in the practical economy essentials of the actual installation. To realize the first, I have quoted from the works of Professors Elmo G. Harris, George Jacob Davis and Carl R. Weidner, and to realize the last, I have incorporated an article by Mr. Arthur H. Diamant together with my own data obtained as before stated.

To thoroughly understand compressed-air pumping, it is necessary that some knowledge of hydraulics and thermodynamics be had. In the later chapters, I have given briefly the

principles of both that should be known. Taken altogether, this work, I think, contains all the information that is necessary to intelligently study, design, install and operate a compressed. pumping plant of any size or capacity.

I desire to express my thanks to the authors from whose works I have quoted and to the various manufacturers named in the text for furnishing the cuts needed.

E. M. I.

June 8, 1914.

PREFACE TO SECOND EDITION

To the First Edition of this work, there has been added some thirty pages of text and eighteen illustrations, together with several formulas and tables. This added matter consists principally of some very reliable and carefully compiled operative data which should prove of interest and value to the student of this subject as well as to the designing and operating engineer.

E. M. I.

Aug. 22, 1919.

CONTENTS

CHAPTER VII

CHAPTER IX

CHAPTER X

CHAPTER XI

CHAPTER XII

CHAPTER XIII

CHAPTER XIV

PUMPING BY COMPRESSED AIR

CHAPTER I

PUMPING WATER BY DIRECT ACTION THROUGH PISTONS

The use of compressed air as an agent for raising and transmitting water and other liquids and semi-liquids is a comparatively new departure; but such rapid strides have been made in its application, and so many systems devised and presented that it now offers one of the most important fields of endeavor for the compressor. The principal advantage derived from the use of compressed air, and the one that makes it so readily adaptable to pumping, is the ease with which it can be transmitted over great distances and the slight losses encountered in so doing. Unlike steam, there are no condensation and but slight radiation losses; in fact the greatest losses are those caused by friction, and they will be discussed at some length in the pages to follow.

Water Horse Power. — To lift water requires the expenditure of work, and the amount of work depends upon the weight of water and the distance it is raised, or

$$Q = q \times H \tag{1}$$

where

Q = work in foot pounds;
q = pounds of water;
H = total lift including friction.

The horse power necessary is expressed by:

$$\text{W.H.P.} = \frac{q \times H}{33{,}000} \tag{2}$$

The value of q in (2) is pounds of water per minute.

The Direct-acting Pump. — In mining, tunneling and other kindred operations, it is often found convenient to use compressed air in place of steam for operating the ordinary direct-acting plunger pump. This type of pump is rugged and reliable, but even when actuated with steam, for which it is designed, it is very uneconomical due to its mechanical construction. There *are no flywheels, and, consequently,* the steam or air must be *admitted during the full length of the* stroke, and at exhaust a cylinder full of air at nearly the initial pressure is discharged into the atmosphere.

Theoretically, a cylinder full of air being used at full pressure is capable of performing the foot pounds of work shown in the following expressions:

$$Q = 144\,(P_1 - P)V_1 \tag{3}$$

where

P_1 = absolute initial pressure in pounds per square inch;
P = absolute final pressure in pounds per square inch;
V_1 = volume of air in cubic feet of compressed air.

Remembering the familiar formula:

$$P_1V_1 = WRT$$

whence

$$V_1 = \frac{WRT}{P_1} = \frac{53.37\,WT_1}{P_1}$$

where

W = weight in pounds of the air;
R = 778 $(C_p - C_v)$ = 53.37 for air;
T_1 = initial absolute temperature in degrees F.

Substituting these values for the equivalents in (3)

$$Q = 144\,(P_1 - P)\,\frac{53.37\,WT_1}{P_1}$$

$$Q = 7685.3\,WT_1\left(1 - \frac{P}{P_1}\right) \tag{4}$$

If the pump is operating at N strokes per minute, *i.e.*, being

supplied with air at the rate of N cylinders full per minute, the horse power would be expressed by:

$$\text{I.H.P.} = \frac{7685.3\, NT_1W\left(1 - \dfrac{P}{P_1}\right)}{33,000}$$

$$\text{I.H.P.} = \frac{NT_1W}{4.3}\left(1 - \frac{P}{P_1}\right) \qquad (5)$$

Formula (5) does not take into consideration the power necessary to overcome the mechanical friction in the pump itself, nor does it take into account the losses due to the excessive clearance in the steam cylinder and ports of the pump. Very large clearance spaces between cylinder heads and piston at the stroke end are provided so as to form a cushion with the contained steam or air and thus eliminate any possibility of the piston striking the heads. These spaces are filled with air at the initial pressure and the air is exhausted into the atmosphere without realizing any return in work from it. The formula just derived then assumes no clearance or friction, and, therefore, is correct only for ideal conditions which mean 100 per cent efficiency.

The average mechanical efficiency of a steam pump is about 80 per cent, that is, 20 per cent of the horse power in the power end is consumed in overcoming friction. The clearance losses will amount to 20 per cent; which means that this amount of air in excess of the volume required by pumping and friction must be provided to replace that lost in the clearance spaces. Therefore, the net available horse power for the actual raising of the water is just 64 per cent of that indicated in (5), or

$$\text{D.H.P.} = 0.15\, NT_1W\left(1 - \frac{P}{P_1}\right) \qquad (6)$$

Obviously, now, formulæ (6) and (2) are equal to each other. The volume of air in cubic feet per minute necessary to perform a certain duty is quite easily determined. When operating at full pressure, it is clear that a volume of compressed air at a pressure

equivalent to the dynamic water-head will lift the same volume of water in cubic feet against that head, or

$$V_1 = v \tag{7}$$

where V_1 = volume of compressed air per minute and v = cubic feet of water per minute to be raised.

This formula assumes, besides no losses, that the air and water cylinders have the same diameter, and is, therefore, adaptable only to the simplest case of direct-pressure pumping. Expressed in cubic feet of free air per minute (7) becomes,

$$V = \frac{P_1}{P} v$$

and for gallons of water per minute instead of cubic feet becomes

$$V = \frac{P_1}{P} \times \frac{(\text{G.P.M.})}{7.481} \tag{8}$$

The actual amount of air may be approximated by adding 20 per cent to the volume obtained from the above formula. The air pressure necessary is found by dividing the total head H in feet by 2.31 and adding 20 per cent to overcome pump friction.

The problem is not always as simple as this; in fact, more often the steam or air cylinder is of greater diameter than the water cylinder. This arrangement permits of the use of air under lower pressure per square inch than that which is equivalent to, or slightly higher than, the dynamic water head. A convenient formula for determining the amount of free air necessary to pump an amount of water against a given pressure is*

$$V = 0.093 \frac{H \times G}{P} \tag{9}$$

Cylinder Proportioning. — Having given the duty and being able either to calculate or to assume other quantities, it is a simple matter to apply the empirical rules of practice and design suitable air and water cylinders to conform to the requirements.

* *Ibid.*

In Table 1 are given the volumes of air in cubic feet per minute necessary to raise water against various heights. The proper cylinder ratios for the various duties are also given.*

Partial Expansion. — If the pump is fitted with a flywheel, crosshead, etc., the air then may be used expansively. That is, the air under full pressure is admitted only during a part of the stroke; is then cut off, and the expansive force of the air completes the stroke. This is plainly a more economical method of operation than the one just described. Referring to Fig. 1, the work done in partial expansion is divided into three parts, namely:

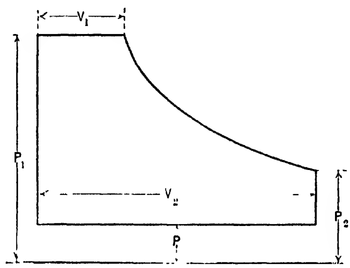


FIG. 1.

1. Work done during the admission of the air:

$$Q_1 = 144\, P_1 V_1$$

2. Work done during the expansion of the air:

$$Q_{11} = 778\, W C_v\, (T_1 - T_2)$$

3. Back-pressure work:

$$Q_{111} = -\, 144\, P V_2$$

The total work done by the air during one stroke of the pump then is the algebraic sum of the three parts, or

$$\begin{aligned} Q &= Q_1 + Q_{11} - Q_{111} \\ &= 144 P_1 V_1 + 778\, W C_v\, (T_1 - T_2) - 144\, P V_2 \qquad (10) \end{aligned}$$

Now

$$V_1 = \frac{WRT_1}{P_1} \quad \text{and} \quad V_2 = \frac{WRT_2}{P_2}$$

$$C_v = 0.169 \quad \text{and} \quad R = 53.37$$

* Laidlow-Dunn-Gordon Catalogue.

TABLE I

Air Consumption of Simple Direct-acting Pumps

Ratios of air to water-cylinder diameters.	0.75-1	0.875-1	1-1	1.125-1	1.25-1	1.5-1	1.75-1	2-1	2.25-1
Air pressure, 100 pounds per square inch									
Head of water against which pump can work	104.0	138.5	185.0	231.5	288.0	416.0	555.0	740.0	925.0
Cubic feet of air used per gallon water pumped	0.675	0.90	1.2	1.5	1.88	2.72	3.60	4.80	6.0
Air pressure, 80 pounds per square inch									
Head of water against which pump can work	83.0	111.0	148.0	181.0	231.0	333.0	444.0	592.0	740.0
Cubic feet of air used per gallon water pumped	0.537	0.72	0.98	1.24	1.55	2.23	2.97	3.96	4.95
Air pressure, 60 pounds per square inch									
Head of water against which pump can work	62.2	83.25	111.0	138.75	173.0	250.0	333.0	444.0	555.0
Cubic feet of air used per gallon water pumped	0.438	0.59	0.78	0.98	1.22	1.765	2.34	3.13	3.91
Air pressure, 50 pounds per square inch									
Head of water against which pump can work	52.0	69.25	97.5	115.75	144.5	207.5	277.5	370.0	462.5
Cubic feet of air used per gallon water pumped	0.368	0.51	0.68	0.85	1.06	1.53	2.04	2.72	3.40
Air pressure, 40 pounds per square inch									
Head of water against which pump can work	41.5	55.5	74.0	60.5	115.5	167.0	222.0	296.0	370.0
Cubic feet of air used per gallon water pumped	0.32	0.45	0.60	0.75	0.895	1.29	1.80	2.40	3.0
Air pressure, 30 pounds per square inch									
Head of water against which pump can work	31.2	41.62	55.5	69.37	86.5	125.0	166.5	222.0	277.5
Cubic feet of air used per gallon water pumped	0.262	0.353	0.47	0.588	0.732	1.05	1.41	1.88	2.35
Air pressure, 20 pounds per square inch									
Head of water against which pump can work	20.75	27.75	37.0	30.25	57.75	87.0	111.0	148.0	185.0
Cubic feet of air used per gallon water pumped	0.204	0.274	0.365	0.457	0.569	0.82	1.09	1.46	1.83

Substituting these values in (10) we have

$$Q = 7685.28\left[T_1 + 0.11\,(T_1 - T_2) - T_1\frac{P}{P_2}\right] \qquad (11)$$

Like formula (3), the foregoing is true only for 100 per cent efficiency and where the clearance losses are zero. The mechanical efficiency and clearance losses of the crank and flywheel pump are approximately the same as those of the direct-acting pump.

By making the simple substitutions necessary in (11), we may now derive an expression of reasonable accuracy showing the available horse power in a certain volume of compressed air for the actual lifting of water when the air is partially expanded in a cylinder of the crank and flywheel pump operating at N strokes per minute.

$$\text{D.H.P.} = \frac{WN}{6.7}\left[T_1 + 2.46\,(T_1 - T_2) - T_1\frac{P}{P_2}\right] \qquad (12)$$

The volume in cubic feet of free air per minute required to raise a given quantity of water against a known dynamic head when employing partial expansion will depend upon the quantity of air admitted to the cylinder, or the cut-off volume. The method of calculation is simple and consists in the substitution of the various values in (10) and solving for V_1. The value of V_2 and T_2 will depend upon the number of expansions employed, which is usually predetermined. The volume V_1 may be then reduced to cubic feet of free air and corrections made for clearance volume in the cylinder.

The initial pressure P_1 is determined from the ratio of air cylinder to water cylinder diameters, and corrections made for frictional losses.

Complete Expansion. — A still more economical method of applying air is to employ complete expansion in the cylinder; that is, allow the contained air after cut-off to expand down to the atmospheric pressure, or nearly so. To obtain a quick disposal of the air at the completion of the stroke of the pump, it is necessary that the exhaust pressure be slightly higher than

the atmospheric. For the sake of discussion and comprehension, we will assume here the theoretical condition, which is that the exhaust pressure is equal to the atmospheric pressure.

The economical advantage of complete expansion over partial expansion lies in the fact that a shorter or earlier cut-off may be employed. For adiabatic expansion, the relation existing between pressures, volumes and temperatures throughout the stroke is quite the same as that for adiabatic compression of air given in Chapter IX. The expression for foot pounds of work is also similar to that for adiabatic compression, with the exception that necessary inversion of some of the values must be made.

The formula is:

$$Q = 498.67\, PV\left[1 - \left(\frac{P}{P_1}\right)^{.29}\right] \tag{13}$$

Correcting for mechanical efficiency and clearance losses (13) becomes

$$Q = 319.15\, PV\left[1 - \left(\frac{P}{P_1}\right)^{.29}\right] \tag{14}$$

The available horse power, then, for lifting the water is expressed by

$$\text{D.H.P.} = 0.0097\, PVN\left[1 - \left(\frac{P}{P_1}\right)^{.29}\right] \tag{15}$$

The volume of free air necessary to perform any given duty may be determined, as before, by substitution in the formulæ and solving for V. In Table 2 are given the cubic feet of free air per minute per I.H.P. required for different cut-offs.

Cylinder Design. — In many instances when the change from steam to air operation is made, the steam pumps already on hand are pressed into service. These pumps are usually designed with a cylinder ratio to take care of an excessive drop in pressure between the boilers and pump. When air is applied, there is a comparatively slight pressure drop and, consequently, the air pressure at the pump throttle valve is far in excess of the steam pressure formerly obtained at that point. The result is that the valve must be partially closed, and heavy losses due to wire-

TABLE 2

CUBIC FEET OF FREE AIR PER MINUTE USED IN A CYLINDER PER I.H.P.

Point of cut-off	Gauge pressure, pounds									
	30	40	50	60	70	80	90	100	110	125
1	23.3	21.3	20.2	19.4	18.8	18.42	18.10	17.8	17.62	17.40
3/4	18.7	17.1	16.1	15.47	15.0	14.6	14.35	14.15	13.98	13.78
2/3	17.85	16.2	15.2	14.5	14.2	13.75	13.47	13.28	13.08	12.90
1/2	16.4	14.5	13.5	12.8	12.3	11.93	11.7	11.48	11.30	11.10
1/3	17.5	15.2	12.9	11.85	11.26	10.8	10.5	10.21	10.02	9.78
1/4	20.6	15.6	13.4	13.3	11.4	10.72	10.31	10.0	9.75	9.42

By F. C. Weber in *Compressed Air*, October, 1896.

drawing friction are imposed. When such a change is made and old pumps must be used, they should at least be rearranged so that the cylinder ratios will be better suited to lifts under which they are to operate. A better method is to replace the old power cylinders with properly designed cylinders in order to meet the various duties.

This cylinder designing is simple and consists merely in the substitution in the formulæ of the number of strokes per minute, initial and final pressures for full pressure, partial or complete expansion operation depending upon the type of pumps, and solving for V_1. This volume is then divided by the strokes per minute to find the volume per stroke. Corrections are next made for the clearance and friction losses before mentioned. Since the stroke length is fixed, the cylinder diameter may be determined by applying the empiric rules.

Compound Pumps. — The final temperatures of the air in the cylinders when cut-off is employed are very low and for this reason the utilization of the expansive force of compressed air was at one time considered impracticable. The theoretical exhaust temperature for any given set of conditions may be computed by substitution in the formula

$$\frac{P}{P_1} = \left(\frac{T}{T_1}\right)^{.29}$$

and solving for T.

The final temperatures met with in practice are higher than the theoretical ones because a certain amount of heat is transmitted to the expanding air from the atmosphere through the cylinder walls and, also, some of the heat generated by the compression of the clearance air at the end of each stroke is absorbed by the incoming air. Moisture carried in the air tends to lower the temperature. When using air at full pressure there occurs also a temperature reduction but this takes place in the exhaust passages and piping because expansion is delayed in this case until the cylinder contents are released.

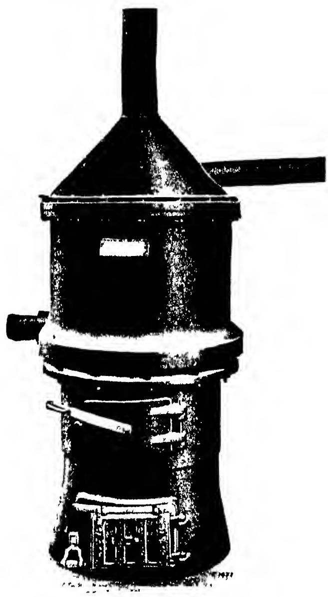


FIG. 2.—Ingersoll-Rand Reheater.

Under ordinary working conditions the exhaust temperature is well below 32° F. when air is used expansively, and often when used at full pressure. The result is that, in the presence of moisture, freezing up of the ports and piping is inevitable and, unless precautions are taken, uninterrupted operation is impossible. The usual means taken to prevent freezing are: withdrawal of the moisture precipitated in the receiver and also near the pump throttle valve and heating the air just before admitting it to the pump cylinder. This latter process is known as reheating and is done to raise the initial temperature of the compressed air to such an extent that the temperature after expansion will be above the freezing point of water. An appliance such as that illustrated in Fig. 2 is employed and is commonly known as a reheater.

Compound pumps have been operated with air used both at full pressure and expansively in each cylinder. For the same *ratios of expansion, the final temperatures of air used expansively in a compound pump are higher than if* but one cylinder were used throughout the range of expansions, but final temperatures are still very low. Unless reheating in some form is employed, the ports and passages in the low-pressure cylinder will become clogged with ice.

Compound pumps have been used in many ways as regards reheating and expanding the air, but by far the most satisfactory method is to furnish cold air to the high-pressure cylinder, exhaust into a reheater where the air is expanded and heated and then convey it to the low-pressure cylinder, using the air at full pressure throughout the stroke in both cylinders.

Return-air System. — In 1891 Mr. Charles Cummings was granted a patent on a "two-pipe system" of operating compressed air engines and pumps. This consisted of an additional pipe line connecting the exhaust of the pump with the compressor intake, and thus forming a closed circuit. By the use of this system the exhaust pressure of the pump, instead of being lost, is utilized in the air-compressor cylinder to increase the initial or intake pressure, thereby necessitating the expenditure of considerably less energy at the compressor. The same air is thus used over and again. Another advantage of the system is that less trouble is encountered in freezing because of the high tension of the enclosed air.

The system is best adapted to pumps operating with full pressure because of the absence of pulsations and consequent uniformity of flow of air to the compressor intake.

The disadvantages are: complications of valves and piping, the high pressures necessary to high efficiency and the first cost. Since the piping necessary is double that used ordinarily, in very remote installations the first cost may become prohibitive; but for short-distance transmission the superior efficiency will undoubtedly overbalance the disadvantage.

The system has been used to some extent in different parts of the country but principally in California and in spite of the fact that it was invented a number of years ago, the essential principles do not seem to be generally understood.

The economical advantages of the return air system are derived largely from the maintenance of high pressures on the enclosed air and thereby providing beneficial compression conditions for the air cylinder. Suppose, for instance, that the

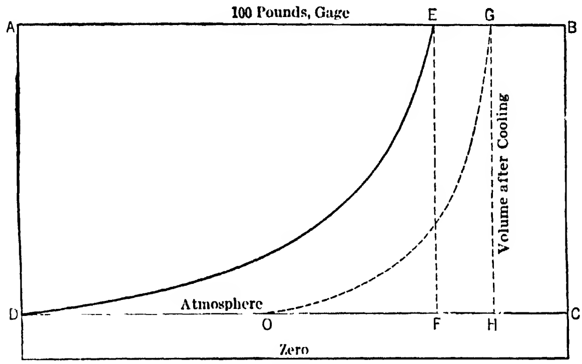


FIG. 3.

requirements of a certain installation were such that an effective pressure of 100 pounds was necessary to pump the liquid. Instead of compressing the air from atmospheric pressure to 100 pounds and, after performing its work, being exhausted into the open as is done in non-return systems, the air in the Cummings' system is compressed from 100 pounds up to 200 pounds (or 200 to 300 pounds) and furnished to the pump at the higher pressure and the pump exhausts against 100 pounds back pressure. Obviously, there is maintained 100 pounds pressure on the air compressor intake.

Less power is required to compress a given volume of free air from 100 pounds to 200 pounds than is necessary to compress the same relative volume from atmospheric pressure to 100 pounds; also, the temperature rise is considerably less and

consequently the power loss due to cooling is minimized. A comparison of the two methods of furnishing the requisite pressure may be made with the aid of Figs. 3 and 4.

Figure 3 illustrates graphically the pressure changes existing inside the air cylinder of the compressor when air is being withdrawn from the atmosphere and compressed to 100 pounds, gauge. *ABCD* is the volume being drawn in during each stroke and compressed adiabatically along the line *DE*. At the end of

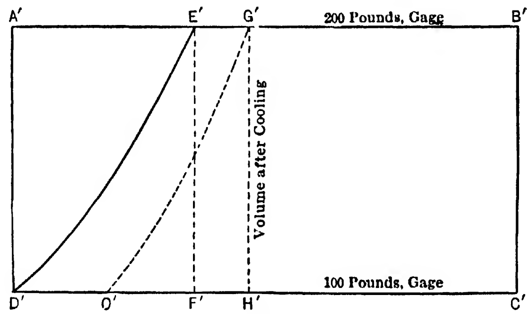


Fig. 4.

compression, the volume has been reduced to *EFCB* and after cooling, further reduced to *GBCH*. Fig. 4 is drawn to the same scale and represents the compression of the same volume (not weight or mass) of air from 100 pounds to 200 pounds pressure. The reduction in volume of the air due to compression and cooling is also shown as in Fig. 3.

The air volume available for work in the pump is shown by *GBCH* in Fig. 3 and by *G′B′C′H′* in Fig. 4 and the former is about one-fourth the size of the latter. The energy expended to compress the air to volume *G′B′H′* is slightly less than double that required to obtain volume *GBCH*, therefore the efficiency of the return air principle is about twice that of the non-return system. The efficiency may be further increased by carrying 200 pounds pressure on the return piping and compressing in the air cylinder from 200 to 300 pounds.

In compressing air from atmospheric pressure to 100 pounds, it is advisable, owing to high temperature rise, to employ a two-stage compressor (see Chapter IX) while on the other hand, in compressing from 100 to 200 pounds, the temperature rise is comparatively slight and a single-stage compressor is suitable. Assumption is made that the initial temperature is the same in both cases.

Steam vs. Air. — Steam and air are alike in that they are both compressible gases, and both follow very closely the laws of sensibly perfect gases. In expanding steam in a cylinder, comparatively slight temperature losses are experienced. The steam cylinders are usually well insulated and every other precaution is taken to preserve the initial temperature. We may say, then, that during the expansion of the steam there is very little change of volume due to the reaction of the reducing temperature on the steam.

In the case of air the conditions are quite different. Very soon after the compressed air leaves the compressor it is cooled down to the temperature of the surrounding atmosphere, and, consequently, the initial temperature of the air in the cylinder of the pump is also equal to that of the atmosphere. When expansion begins, the temperature of the air falls proportionately and the falling temperature reacts to reduce the pressure; and this reduction takes place even more rapidly than the pressure reduction due to the increasing volume.

Comparing the expansion of steam and air, it is easily seen that, with the same initial pressure and the same cut-off, the mean effective or average pressure throughout the stroke of the former is greater than that of the latter. This means that the expansion curve of steam is above the air-expansion curve and, consequently, the steam card is of greater area than the air card. To do the same amount of work with air, then, a much greater volume or initial pressure of air is necessary than to do the same work with steam.

CHAPTER II

THE DISPLACEMENT PUMP

The need of a pump especially designed for use with compressed air and one capable of operation in isolated places without constant attention was recognized by a great many engineers and manufacturers and, consequently, numerous applications for patents on such pumps were made. The majority of the designs presented were impracticable, but the basic principles were similar and the name applied to each was the "Pneumatic Displacement Pump."

In the displacement pump, the compressed air is used to displace the water volume for volume in an enclosed tank or chamber. The air acts as the plunger, exerting the pressure directly upon the surface of the water. The use of pistons, packing glands and other wearing surfaces inside the chamber is thus eliminated, together with the mechanical friction losses so encountered.

Figure 5 is a diagrammatic illustration of a single-tank (or chamber) displacement pump and shows the conditions existing when the discharge of water begins. The chamber has just previously been filled with water by gravity from an outside source and through the check valve indicated by the letter *A*. The compressed air is being admitted to the surface of the enclosed water through the three-way cock *B*. Water is being forced through the check discharge valve *D* and out through the pipe *C* to the point of discharge. When all the contained water has been discharged from the chamber, the three-way cock is shifted, cutting off the live air from the compressor, opening the chamber to the atmosphere and allowing the used air to escape. The weight of the column of water above closes the discharge valve *D* and the weight of the water outside the chamber forces the inlet valve open. As the air is exhausted, the water flows

in, occupying the space just previously filled with air. This *is shown diagrammatically in Fig. 6. When the chamber is filled, the three-way cock is automatically returned to its original* position, and live air is admitted. The inlet valve is now

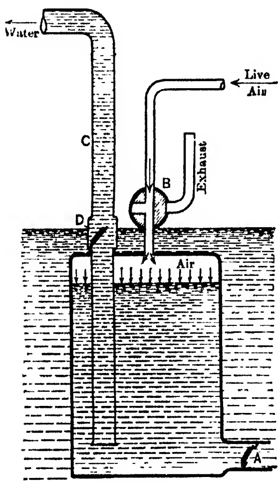


FIG. 5.

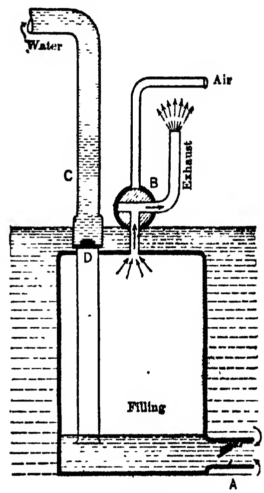


FIG. 6.

closed and the discharge valve opened and pumping begun, and so on.

As in the direct-acting pump, air is used at full pressure in this type of displacement pump and is exhausted into the atmosphere at practically receiver pressure. As before pointed out, this is a wasteful method of using air, but in the displacement pump the total losses are considerably less than in the direct-acting pump. There are no mechanical friction losses to overcome (except the slight friction in the three-way cock and the valves), and the clearance losses are considerably less. Due to the absence of wearing surfaces in contact with the water, the displacement pump can handle with ease quantities of solids, sand and grit as well as acid-laden liquids without injury to itself.

In Fig. 7 is shown an outside view and in Fig. 8 a sectional view of the Halsey single-cylinder displacement pump as formerly manufactured by the Pneumatic Engineering Co. This pump is designed for submergence in the liquid to be pumped. The least allowable depth of submergence is indicated by the horizontal

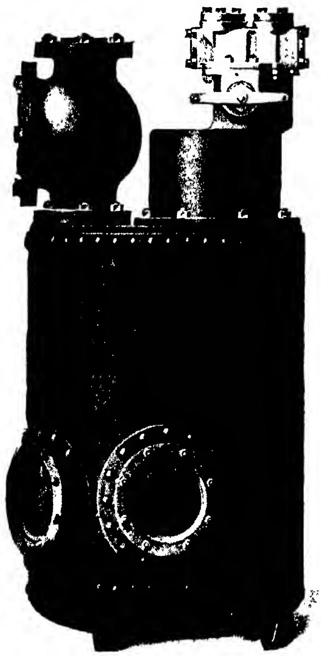


FIG. 7.—Halsey Displacement Pump.

dashed line in Fig. 8. In lieu of submergence the intake of the pump may be connected by piping to an otherwise isolated water head that is equivalent to the depth of submergence required.

As shown, the Halsey pump consists of a steel-riveted chamber, on the head of which is mounted two castings; one containing a piston air valve and operating mechanism, and the other con-

taining a ball-check discharge valve. Inside the chamber is a float which rides on the rising and falling water surface and over a rod which is connected to the air-valve operating mechanism. Extending down from the discharge valve is a pipe of equal diameter of discharge pipe and ending in a bell mouth near the

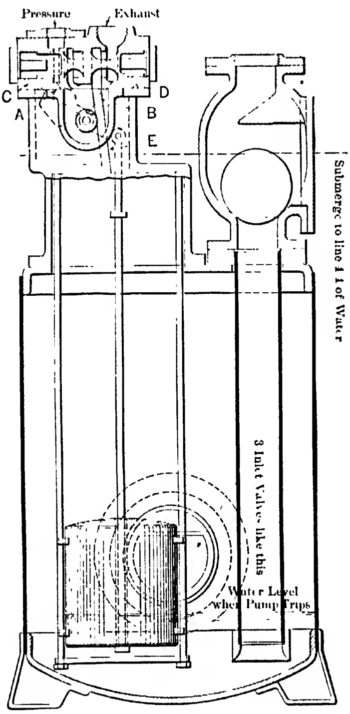


FIG. 8.

bottom of the chamber. The operation is as follows: Water enters the chamber through the swing-check foot valves, and the level rises inside, carrying the float. When the chamber is filled the float engages the upper collar shown on the rod and shifts the air valve. The exhaust port is thus closed, compressed air is admitted to the water surface and pumping begins. As the

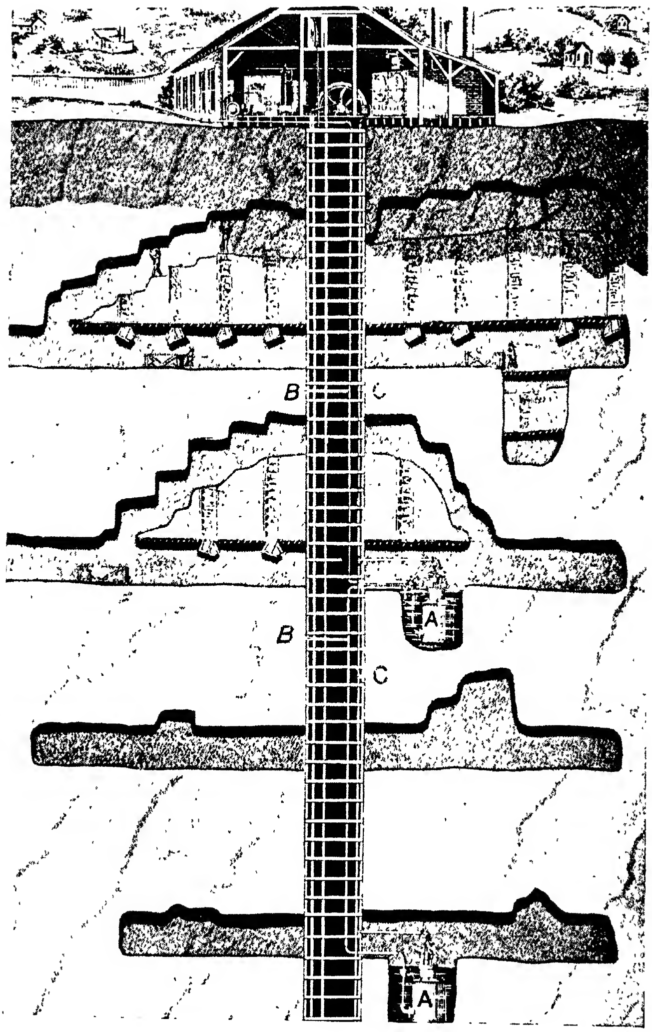


FIG. 9.—Halsey Pump Installation in a Mine.

surface of the water is lowered, the float follows until the chamber is empty, when the float engages the lower collar on the rod and the air valve is shifted to its original position. This shift cuts off the compressed air, opens the exhaust port and the used air escapes. The chamber again fills and the operation is repeated.

In Fig. 9 is shown a pair of Halsey pumps installed at different depths in a mine. Both pumps receive their air from a common pipe and also discharge into the same flow pipe.

Figure 10 illustrates an installation of a Weber single-cylinder displacement pump of the Suburban type which is especially designed for operation in drilled wells. There are but two moving parts within the pump itself and these are bronze ball valves for inlet and discharge of water. Air is admitted and exhausted by the operation of a piston controlling valve which requires no oiling or special attention and which may be mounted in any convenient place above ground.

The outfit shown is entirely autmoatic and water is instantly available at all times. The electric motor, which may be either belted or geared to the compressor, is equipped with a pressure control switch which starts and stops the motor when predetermined air pressures are reached in the receiver. The operation of the system is as follows: The motor is started and the pressure in the receiver increases up to the point for which the control switch is set to break the circuit when immediately the motor is stopped. The system is now charged and water may be drawn at any point in the discharge piping. When water is being withdrawn, the pump is in operation, consuming the air in the receiver and piping and which obviously causes a reduction in air pressure. When, by continued operation of the pump, the air pressure has been lowered to a point slightly higher than that actually required to lift the water the necessary height, the control switch automatically starts the motor and the system is re-charged.

When electric current is not available, a gasoline or steam engine may be substituted for the motor and in which case, it

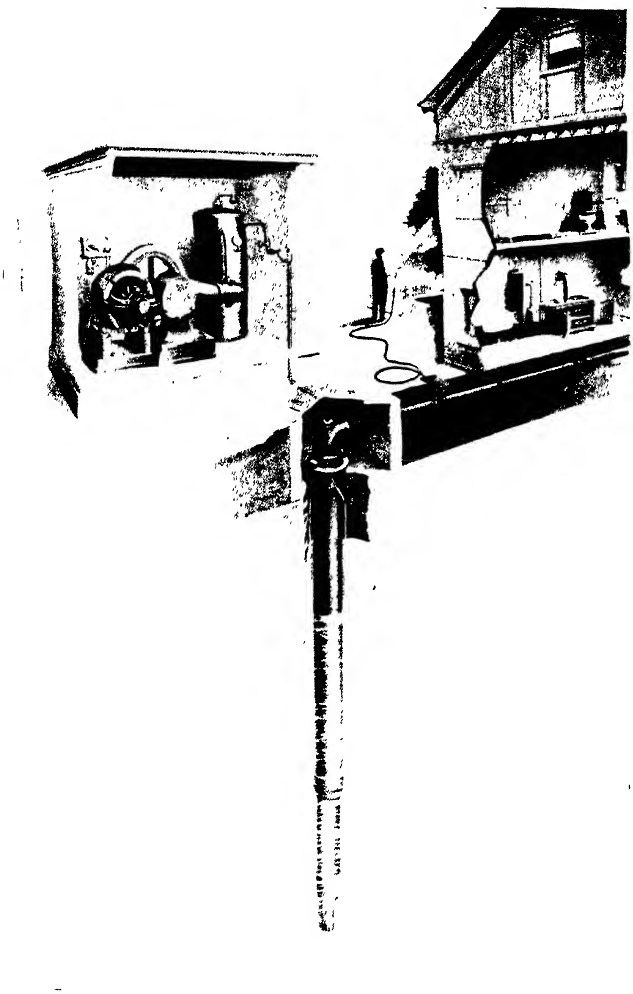


FIG. 10.

becomes necessary to start and stop by hand whenever the storage is to be replenished or is sufficient.

The system may be used to advantage on suburban estates, clubs and other places where relatively small quantities of water under medium pressures are required.

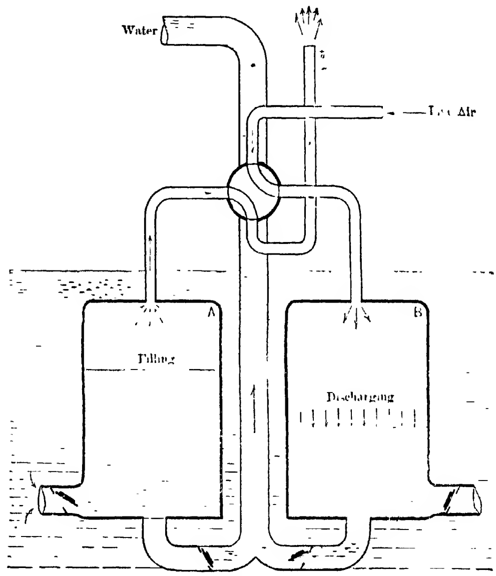


FIG. 11.

The flow of water from the single-chamber type of pump is intermittent, for during the period of time required to exhaust the air and fill the chamber with water, the discharging has ceased. In the twin-chamber displacement pump this objection is almost entirely removed; for while one chamber is discharging, the other one is being filled, and, consequently, the only time lost to actual

pumping is the short period required by the valve mechanism in moving the air valve. Fig. 11 is a diagrammatic illustration of a pumping apparatus of this type. The four-way valve shown mounted above the chamber heads controls the live air and "switches" it from chamber to chamber at the proper moment. The exhaust air from each chamber is also regulated by the same valve.

Figure 12 illustrates the conditions that exist during the time that chamber *A* is being emptied of its water and chamber *B*

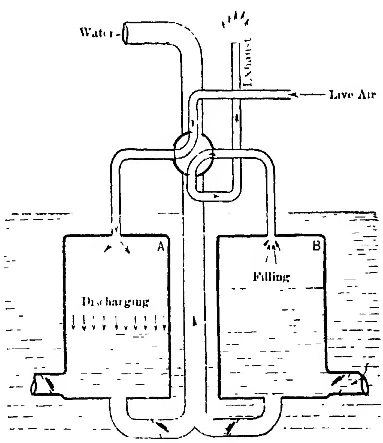


FIG. 12.

is being filled. The one is emptied in a slightly greater length of time than is required to fill the other. Immediately after *A* is emptied of water the four-way valve is automatically shifted, and the compressed air from the compressor is admitted to *B*, and the used air in *A* allowed to escape to the atmosphere. These conditions are illustrated in Fig. 11. When the water in *B* has been pumped, *A* has been filled and the valve is again shifted, and *A* is emptied and *B* filled with water, and so on.

Each chamber is provided with check inlet and discharge valves which operate in the same manner as those of the single-chamber pump previously discussed.

In Figs. 13 and 14 are shown outside and sectional views of the Latta-Martin twin chamber displacement pump. This pump is also designed either for submergence in the liquid to be pumped or else connected by piping to an outside water head of sufficient height to permit rapid filling of the chambers.

FIG. 13. — The Latta-Martin Pneumatic Displacement Pump.

As shown, in each chamber there is provided a small copper float suspended on a pipe connected to the auxiliary valve above. Housed in suitable castings and mounted on top and across the chamber heads are two piston-controlling valves. The smaller one moves over ports connected with the main valve, and the arrangement is such that the small auxiliary valve (which is controlled by the float) controls the movements of the main valve which, in turn, controls the live-air supply. The operation of the pump is as follows:

When the water level has been lowered in one tank to a point just above the discharge opening, the small copper float drops

by gravity a distance of about one quarter of an inch. The falling float carries the lever to which it is attached, and a small port is opened into which enters some of the surrounding air held in the chamber. The air passes up through the float pipe and into passages in the valve casting above which lead to the end of the small auxiliary valve. The small auxiliary valve is

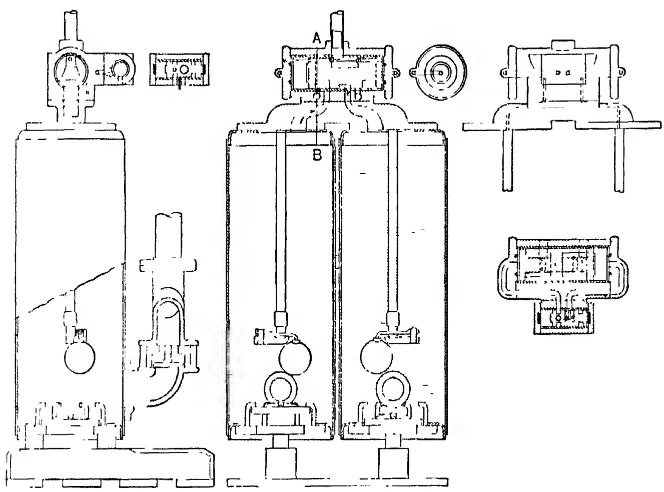


FIG. 14.

thus shifted to the other end of its travel by the air pressure and ports leading to the main valve are uncovered. This admits air pressure to main valve. The main valve is now moved and a passageway connecting the live air supply and the pump chamber is opened. Live air is now admitted to the chamber that has been filled with water and the other chamber is exhausted of the used air. After emptying the second chamber the other is filled with water and the valve movement is reversed and so on. In Fig. 15 is a typical Latta-Martin pump installation.

Figure 16 is a section of the Shone Pneumatic Sewage Ejector. As shown, it consists of an enclosed chamber provided with

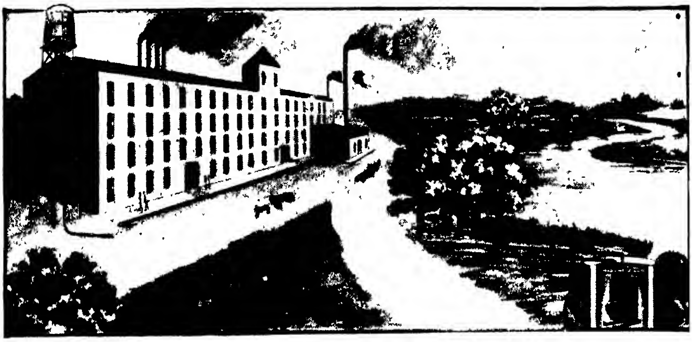


FIG. 15.—Typical Latta-Martin Pump Installation.

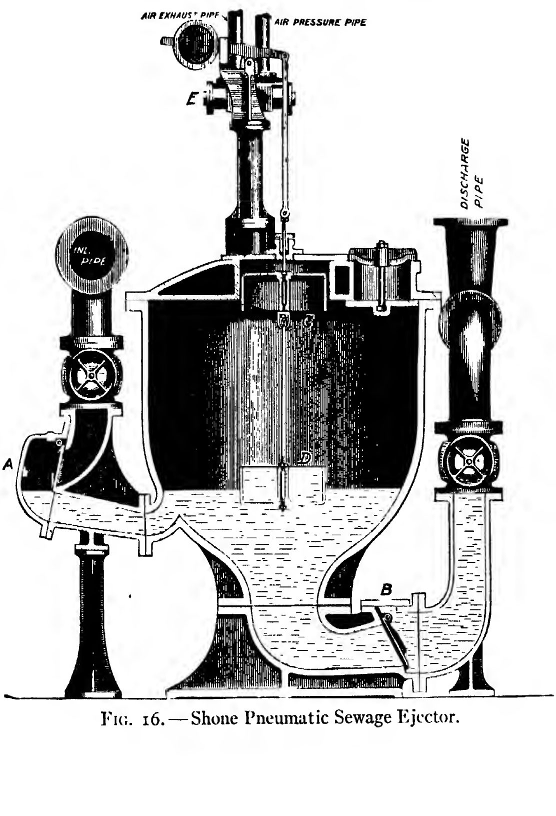


FIG. 16.—Shone Pneumatic Sewage Ejector.

suitable sewage inlet and discharge connections, together with their check valves. Inside the chamber are two cast-iron bells linked together. The upper bell is connected to the automatic valve by a rod passing through a stuffing box. The operation of this pump is as follows:

When the level of the sewage has been lowered to the point shown in the illustration, the weight of the lower bell and its contents moves the valve and the live air is cut off and the exhaust opened. The inlet valve is then opened by the weight of the fluid column and inflow of sewage begun. When the level has reached the upper bell, air is enclosed and the level continues to rise around the bell until the buoyancy is sufficient to raise the lower bell with the rod and the air valve is shifted to its first position. This closes the exhaust port and admits compressed air, when the chamber contents are again pumped out. These ejectors are usually installed in pairs giving in effect a double-chamber unit. A typical installation is shown in Fig. 17.

Advantages.—The advantages claimed by displacement pump manufacturers are briefly the following:

1. No close fitting or wearing surfaces in the chambers.
2. No piston leakage, no mechanical friction and small clearance losses.
3. No changes or adjustments necessary to adapt the pump to varying conditions or lifts.
4. The ability of the pump to operate when submerged in the liquid and its facility for handling large percentages of solids and grit.
5. Large capacities.
6. Automatic control and the consequent impossibility of racing, and little attention necessary by the operator.

Air Consumption. — The volume of compressed air necessary to operate a displacement pump, neglecting clearance, is equal to the volume of water to be pumped, or $V = v$, where V = cubic feet of compressed air per minute and v = cubic feet of water

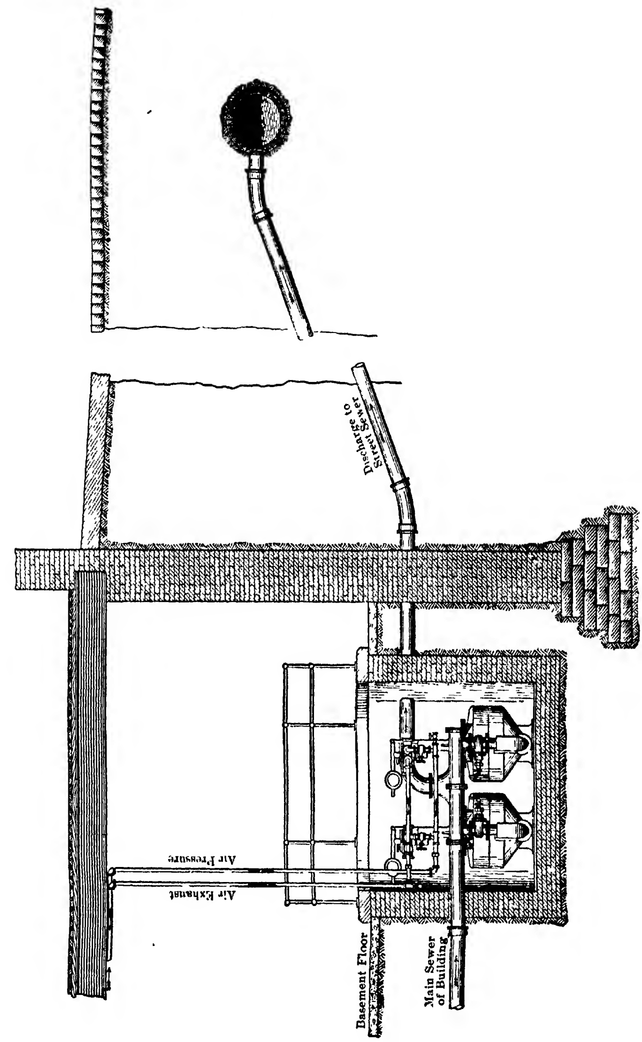

Discharge to
Street Sewer
Air Pressure
Air Exhaust
Basement Floor
Main Sewer
of Building

FIG. 17.

delivered per minute. Expressed in free air per minute, the expression becomes

$$\frac{P}{P_1}V = V_1 = \frac{P}{P_1}v \tag{16}$$

where P is equal to the pressure pumped against and P_1 is equal to 14.7 pounds or the atmospheric pressure.

There is a loss of about 10 per cent due to clearance in the pump and hence, in applying the above formula, correction to this amount should be made. The pressure P is found by dividing the total dynamic water head in feet by 2.31 and adding 14.7 pounds.

To illustrate the method of application of the formulæ given, assume a set of conditions as follows:

With a total dynamic head of 100 feet, how many cubic feet of free air per minute are necessary to lift 500 gallons of water per minute?

$$\text{Pressure} = \frac{100}{2.31} = 43.4 \text{ pounds gauge}$$

or 58.1 pounds absolute.

$$\text{Cubic feet of water per minute} = \frac{500}{7.481} = 67.$$

Substituting in formula:

$$V_1 = \frac{58}{14.7} \times 67 = 260 \text{ cubic feet of free air per minute.}$$

To the air volume just found must be added 10 per cent to cover clearance losses, and to the pressure must be added the friction loss in the air line connecting the compressor and the pump. Also if piston-displacement volume is required to be found, a further correction of air volume is to be made for the volumetric efficiency of the compressor.

Table 3 gives the cubic feet of free air per minute required to lift one gallon of water per minute against various pressures. The air volumes given are actual and a compressor capable of delivering the net amounts of air should be chosen.

TABLE 3

CUBIC FEET FREE AIR REQUIRED PER GALLON OF WATER AT VARIOUS PRESSURES FROM 5 TO 150 POUNDS PER SQUARE INCH

Gauge pressure, pounds	Cubic feet free air required per gallon water	Gauge pressure, pounds	Cubic feet free air required per gallon water
5	0 179	80	0.861
10	0 224	85	0 906
15	0 270	90	0.952
20	0 315	95	0 997
25	0.361	100	1.043
30	0.406	105	1.088
35	0 452	110	1.134
40	0.497	115	1.179
45	0.543	120	1.225
50	0.588	125	1.270
55	0.634	130	1.316
60	0.679	135	1.361
65	0.724	140	1.407
70	0.770	145	1.452
75	0.815	150	1.498

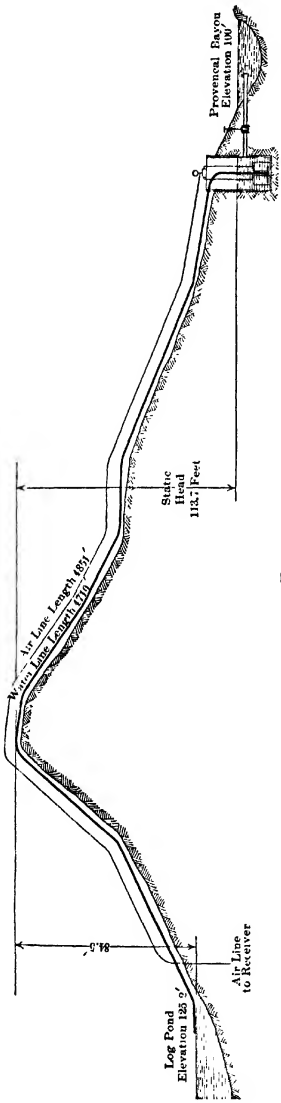


Fig. 18.

Performance of a Displacement Pump. — Comparatively few dependable tests of the displacement pump have been made by disinterested engineers, and this may be accounted for by the fact that the system is so far superior to other methods of distance pumping that exhaustive tests are not necessary to demonstrate the advantages. Then again, the system is new and competition of the few manufacturers has not become sufficiently keen to necessitate duty guarantees and comparative tests On the following pages are given the results of one test made by the writer on a displacement pump installed under rather severe conditions. These results with the explanatory remarks will give a good idea of what may be expected from the system in the way of operating efficiency under similar conditions.

General Remarks.—The source of water supply of the saw mill of the Louisiana Long Leaf Lumber Co. at Victoria, La., was a bayou located about a mile away from the engine room and log pond. Prior to 1908, water was furnished by a direct-acting steam pump lo-

cated, together with a donkey boiler, on the bayou bank. This installation necessitated not only a day and a night man in attendance at the pump house, but also kept a team busy hauling wood for the boiler. In 1908 an Ingersoll-Rand compressor and an Allison displacement pump were installed. Fig. 18 is a diagrammatic illustration of the completed installation.

Equipment. — The machinery installed consisted of the following:

1 6 in. by 6 in. by 6 in. Class R. C. steam-driven compressor:	
Maker	Ingersoll-Rand Company
Rated speed in r.p.m.	185
Piston displacement per minute in cubic feet	35
Rated pressure in pounds	100
1 No. 0 air receiver:	
Maker	Ingersoll-Rand Company
Length in feet	6
Diameter in inches	18

The receiver was fitted with gauge, safety valve and blow-off valve and nipple.

1 Allison displacement pump:	
Maker	Harris Air Pump Co.
Diameters of cylinders in inches	18
Length of cylinders in inches	18
Rated capacity in G.P.M.	25–30

Figures 19 and 20 show the types of compressor and receiver used.

The conditions under which the pump operated were the following:

Length of water line in feet	4710
Diameter of water line in inches	3
Length of air line in feet	4851
Diameter of air line in inches	$1\frac{1}{2}$
Static pumping head in feet	113.7

Method of Test Procedure. — A number of trial runs were made as is usual in tests of this kind. During each trial, readings were made of the receiver gauges, boiler gauge and thermometer. The compressor revolutions were counted and diagrams were

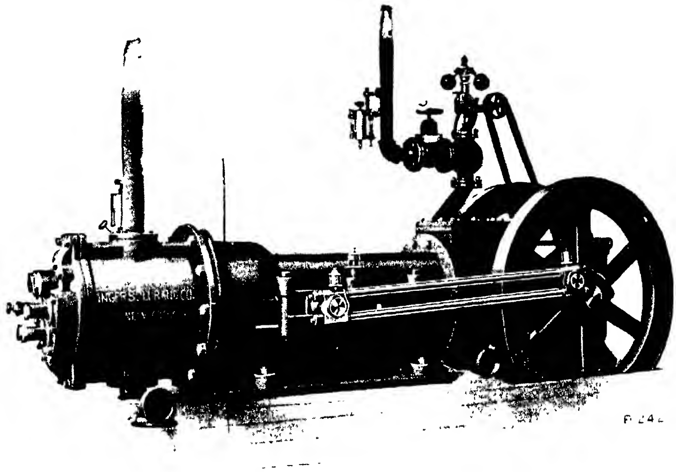


FIG. 19. — Ingersoll-Rand Class R C. Compressor.

taken from the air cylinder with a Robertson-Thomson indicator fitted with a 40-pound spring.

Computations. — The friction loss in the air line was found by placing a gauge in the air line at the pump and subtracting this gauge reading from that of the receiver gauge. Both gauges were new and assumed to be accurate.

The friction loss in the water line was obtained by subtracting the actual measured static head from the pressure head indicated by the gauge at the pump. This head difference included loss due to the resistance of the pump discharge valves and connections. The air horse power and volumetric efficiency were determined from the indicator diagrams in the usual manner. The water horse power was determined by multiplying the weight of water pumped per minute by the dynamic head and dividing the product by 33,000.

The free air consumption in cubic feet per minute was computed by multiplying the cubic feet of piston displacement per minute by the volumetric efficiency. Corrections were made for intake temperature as read from the thermometer. The water pumped was measured by timing the flow in a barrel of known cubical contents. This was easily and accurately accomplished owing to the small quantity being handled.

FIG. 20. — Ingersoll-Rand Receiver.

Results. — Table 4 is a log of results of the test together with the computations made therefrom. The efficiencies obtained are quite low and show a wide variation with the theoretical. Disregarding clearance and leakage and neglecting the air necessary to operate the controlling valve (which in this case was operated with live air), the theoretical efficiency would amount to approximately 35 per cent. The difference between the theoretical efficiency and the actual efficiency of the system found shows the waste of leakage and clearance and the power required to operate the controlling valve.

The installation of the air equipment in place of the steam pump proved to be a wise move. The cost of the compressor pump, piping, installation and labor amounted to $1284.00. The old water line was used and the piping cost mentioned was the purchase and laying of the 1½-inch air line. With the compressor located in the engine room, and under the supervision of the regular operating force, and since the fuel cost was nothing, the operating expenses of the system were only the amount paid for oils, waste, packing, etc. The cost of operating the old steam

TABLE 4

TEST OF A DISPLACEMENT PUMP

(Oct. 20, 1908)

Number	R. p. m.	Piston displacement in cubic feet per minute	Temperature in degrees F.	Volumetric efficiency	Actual cubic feet of free air per minute delivered	Gallons of water per minute pumped	Air Pressures At receiver	Air Pressures At pump	Friction loss in air line in pounds	Friction loss in water line in feet	Total pumping head in feet	Water horse power	Air horse power at compressor	Air horse power at pump	Efficiency of the system	Efficiency of the pump	Boiler pressures
1	175	33.25	60	85	28.3	25.6	85	80	5	71.1	184.8	1.195	5.0	4.7	24.0	25.5	125
2	176	33.44	61	85	28.4	25.6	85	80	5	71.1	184.8	1.195	5.1	4.8	23.6	25.0	125
3	175	33.25	62	84	27.97	25.6	85	80	5	71.1	184.8	1.195	5.0	4.7	24.0	25.5	125
4	178	33.82	60	85	28.69	26.0	85	80	5	71.1	184.8	1.2	5.1	4.8	23.6	25.0	130

pump, including wages of attendants and hauling of fuel, averaged $110.00 per month. This is exclusive of the supplies, such as oil, waste, and packing. The saving realized by the pneumatic equipment amounts to $1310.00 per year, and, consequently, the system pays for itself each year of operation.

While all conditions may not be quite so favorable as these to the installation of the displacement pump, still there are few instances of long-distance pumping where the displacement

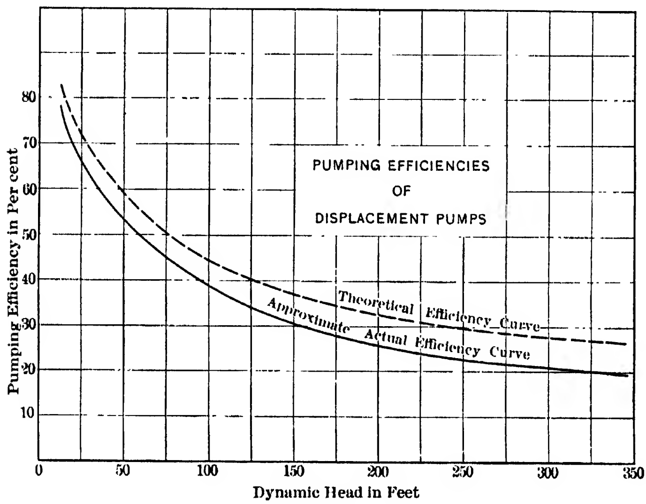


FIG. 21.

pump in some form will not show a high rate of interest on the amount of money so invested.

Efficiency. — In Fig. 21 are plotted two efficiency curves. The ordinate represents efficiency in per cent, and the abscissa represents feet of lift. The dotted curve is theoretical pumping efficiency and the solid curve is the actual efficiency plotted from average data furnished by several manufacturers of the displacement pump and from tests made by the writer.

An examination of these curves shows that the efficiency falls rapidly with increase of pumping head from 10 to 100 feet and

ılls gradually on further head increase. Therefore, it is evident that to realize the highest efficiency, pumping in stages is necessary, and the more stages employed the higher will be the efficiency. Fig. 22 illustrates quite clearly the meaning of stage pumping.

The number of stages to be employed depends largely upon local conditions. It would have been unwise, for instance, to have installed several pumps at the Louisiana Long Leaf Lumber

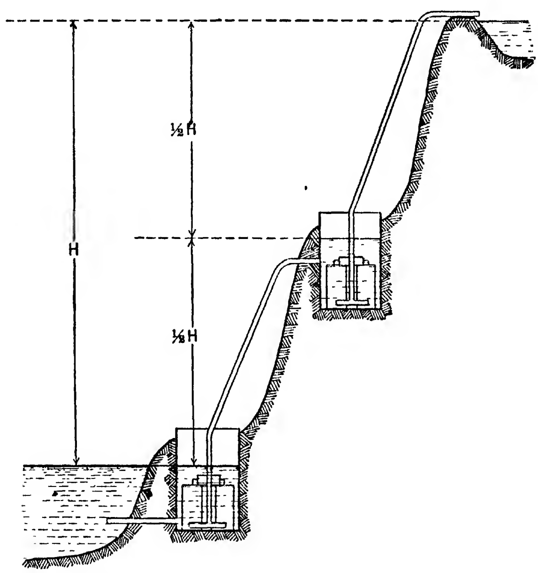


FIG. 22.

Co.'s mill because the quantity of water needed was small and the fuel cost *nil.* Other conditions limiting the number of stages are first cost, interest, depreciation and complications of the system.

Still another consideration that must be taken into account is that, in stage pumping where the air is being furnished to all pumps by one supply pipe, throttling losses occur unless great

care is used to divide the total head equally among the stages. A good rule to follow is to use one stage for each seventy-five feet of dynamic head. Under this head about 45 per cent efficiency can be looked for, and has been actually obtained in several instances.

Design. — A properly designed mechanical installation of any sort is that combination of apparatus, the sum of whose operating cost, interest and depreciation on the first cost, upkeep and complications is least for the conditions at hand. This applies everywhere, and to no appliances more fittingly than to pumping plants. The method to employ in designing a displacement pumping system is quite similar to that employed in the air-line design explained in a succeeding chapter. The operating cost plus interest and depreciation charges is estimated for single- and for multi-stage installations and the results are compared. That which shows the least cost, or, in other words, best over-all efficiency is the design to adopt. After deciding on the apparatus, the water and air lines are designed in accordance with principles given in a later chapter.

CHAPTER III

RETURN-AIR SYSTEM

When the water has been forced from the cylinder of the displacement pump, the air volume remaining is allowed to escape into the atmosphere through the controlling valve. This air is at practically initial pressure and, consequently, the energy of expansion in the air is lost. If the exhaust from each chamber is piped back to the compressor intake, the entire system would be closed to the atmosphere, and the force of expansion in the air after displacement of the water in the pump chamber is exerted against the compressor piston on the suction side and thus assists in the compression of the air on the reverse side of the piston. This is the basic principle of the return-air system and, clearly, the operation is decidedly more economical than that of the plain displacement pumping systems.

Principle. — Figs. 23 and 24 are diagrams of a return-air system with submerged tanks. Referring to Fig. 23, H is the air cylinder of the compressor; J is an automatic compensating valve whose duty it is to replace any air that may from time to time be lost by leakage, absorption or in switch operation; F is the automatic switch which controls the live and exhaust air to and from the tanks; A' and B' are the air pipes connecting the tanks and the switch; E is the water discharge pipe, which is connected by branch pipes to the tank riser pipes G_1 and G_2; A and B are the tanks, each fitted with check inlet and discharge valves similar to the plain displacement pump. The operation is as follows:

In the diagram, Fig. 23, tank B has just been emptied of water and the used air is being drawn intó the compressor cylinder through the pipe B'. As the air is thus being drawn into the compressor cylinder, the tank B is filling through the check valve

C 2′ and at the same time live air is being forced through pipe A' into tank A, forcing the water out the discharge pipe E. When tank A has been emptied, tank B is full and the switch is automatically shifted and live air is admitted to tank B and the high-pressure air in tank A is returned to the compressor cylinder through the pipe A', and so on.

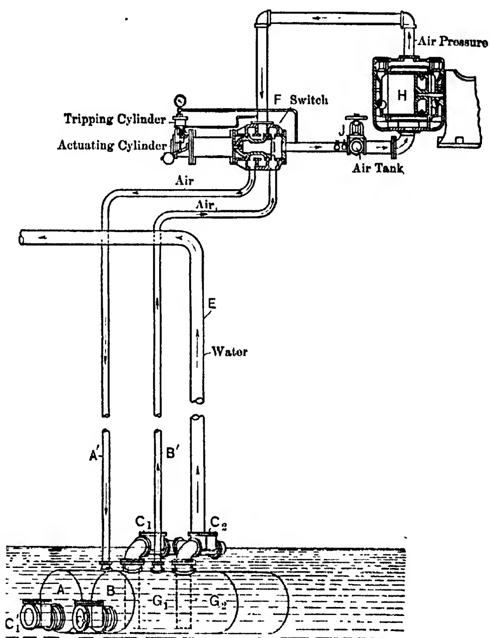


FIG. 23.

The air as it is exhausted from each pump chamber expands back through the air line and switch into the air cylinder through the inlet valves and out of the compressor cylinder through the discharge valves. It continues on through the other end of the switch down the other air line into the other chamber which has been filled with water. The compressor is operating all the while, but doing no work other than overcoming the mechanical

friction of itself because the pressures are equal on both sides of the piston. The rush of air continues until the pressure throughout the system is equalized when, immediately, the com-

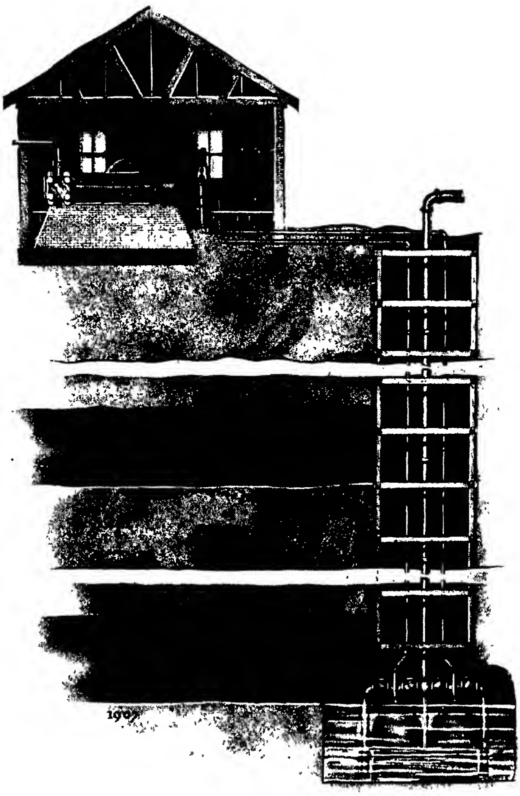


Fig. 24.

pressor takes up its load, compressing and furnishing air to the filled tank, and drawing air from the empty tank. When the air pressure in the empty tank is reduced below that of the water head outside, then inflow begins through the inlet check valve.

The chambers are not necessarily submerged because as the compressor continues to operate withdrawing air, a partial vacuum will be created in the air line and tank and water will be lifted by vacuum, filling the tank.

Switch.--There are two types of switches that may be employed in connection with the system, *i.e.*, the automatic and the mechanical. The choice of any type depends upon the

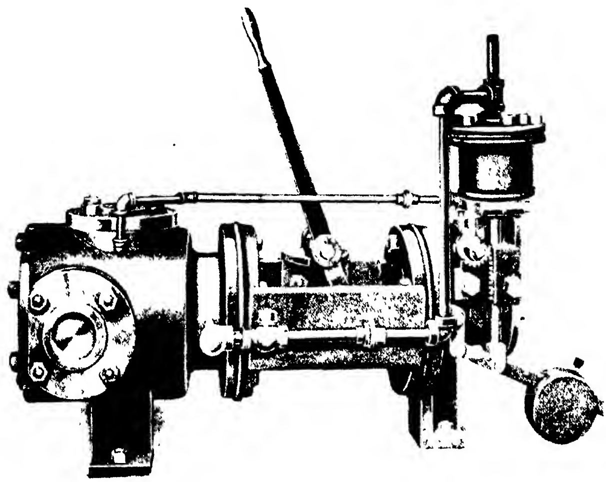


FIG. 25.

conditions. The automatic type is operated or thrown by the difference in air pressures inside and outside the system. The device consists merely of a piston valve and each throw corresponds to the filling or emptying of one of the tanks. Referring to Fig. 23, the operation of the automatic switch is as follows: Equilibrium of pressures has been established throughout the system, and tank *B* has been filled with water by gravity. As the compressor continues to operate, a partial vacuum is created in the piping above the water level in the filled tank, and on one

end of the piston valve. The other end of the valve is exposed to atmospheric pressure, and, consequently, the pressure difference on the ends shifts the valve. When the other tank is filled, the cycle is repeated but reverse in action. The valve may be adjusted to operate on any required vacuum; so that if it is necessary to operate with a suction lift to the tanks, a correspondingly higher vacuum may be provided. In Fig. 25 is an illustration of this switch.

The mechanical switch is merely an appliance which is set to operate at a predetermined number of revolutions of the compressor. The required number of revolutions is determined by computing the volume of air necessary to completely discharge one of the tanks. These computations can be made very accurately beforehand, but a check test is usually made after installation of the equipment.

Compensating Valve. — After the system is in operation, there will be certain losses of air due to leakage through imperfect joints in the piping, due to absorption of the air by the water; and a small volume of air is consumed in shifting the automatic controlling valve. This lost air must be replaced by air from the atmosphere, otherwise the system in time will become inoperative. The loss is replaced by a compensating valve (Fig. 23) placed in the compressor suction pipe between the switch and the air cylinder. This valve is merely an atmospheric check which opens when the pressure in the pipe drops below the permissible vacuum. A globe valve is used in connection with the automatic valve. This globe valve is placed outside the compensating valve and is so adjusted that the proper amount of free air may be admitted while the system is in operation.

Starting. — When the return-air plant is first started it is necessary to first charge the system with air drawn from the atmosphere. The first cycle of operation then is exactly the same as that of the ordinary displacement pump previously discussed, and free air is drawn into the compressor cylinder

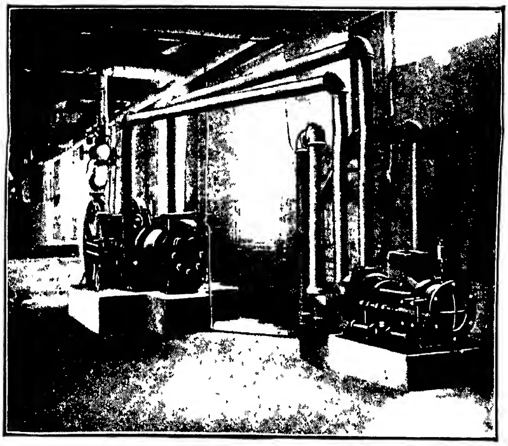


Fig. 26.

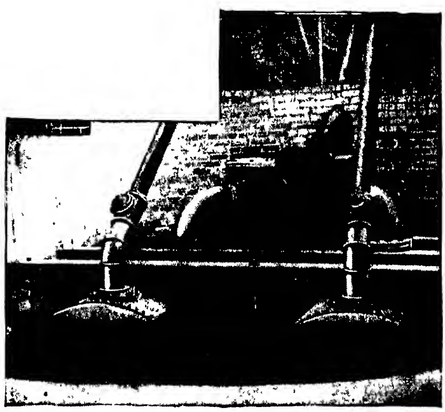


Fig. 27.

through the compensating and globe valves. After the required pressure has been established throughout the system, the compensating valve closes automatically and the return-air principle begins.

In Figs. 26 and 27 are shown two views of a return-air system installed at the plant of the Holland Sugar Co. The capacity of this plant was 2,350,000 gallons of water per day of twenty-four hours and the lift, 70 feet.

Proportioning. — Professor Elmo G. Harris in a discussion published in Volume LIV of the *American Society of Civil Engineers* has prepared a mathematical analysis of the return-air system. This analysis is given in full on the following pages.

Harris Theory. — "With the development of this system of pumping, many problems have been presented for solution, some purely mechanical, while others require a mathematical analysis. The latter have proved very interesting and instructive.

"In the process of such analysis, it will be necessary to use the following symbols. Though the analysis may be considered intricate, the final formulæ are unexpectedly simple and easy of application.

"Let P_o = Delivery pressure — a constant — in pounds per square inch;

P_1 = Pressure throughout the system immediately after switching;

P_x = Pressure of air entering compressor — a variable;

V = Volume of one pump tank — a constant — in cubic feet;

V_y = Volume of air in delivering tank at pressure P_o — a variable;

nV = Volume of one air pipe;

p_1 = Pressure at which water begins to enter tank from which air is being exhausted;

p_o = Lowest pressure reached (this occurs just before switching);

q_a = Effective volume, intake of compressor, in cubic feet per second;

q_w = Average water delivery, in cubic feet per second;
Q = Total volume taken into compressor, while working pressure down from P_1 to p_1, or approximately P_o to p_1 in any case and approximately P_o to p_o when tanks are near surface of water supply;

R_o = ratio $\frac{P_o}{p_o}$;

R_1 = ratio $\frac{P_1}{p_1}$.

"All pressures are 'absolute,' that is, gauge pressure + 14.7 pounds.

"**Compressor Capacity** (= q_a). — The first problem is to find the necessary intake capacity of the compressor. In this, fortunately, the problems of work and temperature inside the compressor need not be considered, and, therefore, in the analysis, the temperature of the air may be considered as constant, though it will be necessary, finally, to apply a coefficient to provide for the effect of expansion due to the heating of the air as it passes through the hot intake valves.

"Assume that a small volume dQ of air at the pressure P_x is taken out of the exhausting tank and forced into the delivery tank, where the pressure is P_o, and its volume is dV_y, then, by the law that the pressure multiplied by the volume is constant:

$$P_x\,dQ = P_o\,dV_y;\ \text{or}\ dQ = \frac{P_o}{P_x}dV_y \tag{17}$$

"Also, by the same law, the sum of the product of the pressure multiplied by the volume must be constant, since the quantity (or mass) of air in the system does not change. When one tank is full of water, and its air pipe is full of air at the pressure, p_o, the other tank and air pipe must be full of air at the pressure, P_o. Under this condition, the sum of the products is

$$P_oV(1+n) + p_oVn$$

At any other time the sum of the product is

$$P_xV(1+n) + P_o(V_y + nV)$$

"Hence,

$$P_oV(1+n) + p_onV = P_xV(1+n) + P_o(V_v + nV) \quad (18)$$

"To simplify, put $p_o = \frac{P_o}{R_o}$ and equation (18) reduces to

$$\frac{P_o}{P_x} = \frac{V(1+n)}{V\left(1+\frac{n}{R_o}\right) - V_v} \quad (19)$$

"Substitute equation (19) in equation (17), and

$$dQ = V(1+n)\frac{dV_v}{V\left(1+\frac{n}{R_o}\right) - V_v}$$

"Integrating between the limits, $V_v = V_1$ and $V_v = 0$, there results:

$$Q = V(1+n)\log_e\frac{V\left(1+\frac{n}{R_o}\right)}{V\left(1+\frac{n}{R_o}\right) - V_1} \quad (20)$$

"Let V_1 represent the volume of air in the delivery, or high-pressure tank, when water begins to enter the other; that is, when the pressure in the other tank has dropped to p_1; this marks a change in the operation; see Fig. 28. Just at this period there must be enough air, at the pressure p_1 in the volume $V(1+n)$, to fill the space $V - V_1$, at the pressure P_o, in the other tank, and its own air pipe at the pressure p_o. Hence the equation:

$$p_1V(1+n) = P_o(V - V_1) + p_onV \quad (21)$$

or

$$P_oV_1 = V[P_o - p_1 + n(p_o - p_1)]$$

"Now, n is a fraction, and p_o and p_1 are small and nearly equal, in practice; hence $n(p_o - p_1)$ can be neglected. Then:

$$V_1 = \frac{V}{P_o}(P_o - p_1) \quad (22)$$

PROPORTIONS FOR A COMPOUND DIRECT-AIR-PRESSURE PUMP.

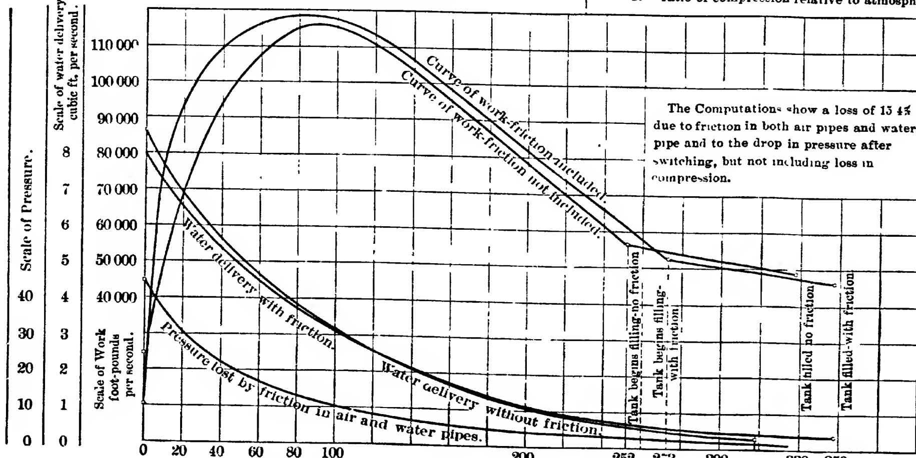
Requirements·
Lift 1 000 gal. per min. (= 2.23 cu. ft. per sec.) through 50 ft., vertical.
Length of both air pipes and of water pipe each being 600 ft.

Proportions·
Compressor displacement, 9.5 cu. ft. per sec.
Air pipes 5 in. diameter.
Pump tanks 816 cu. ft.
Water pipe 12 in. (may be reduced to 10 in. with but little loss.)
Time between switchings 360 sec. = 6 min.

Formula used to get pressure loss in friction in air pipes.

$$p = 0.000002\,\frac{l}{d}\,v^2 R$$

p = pressure loss, in pounds per square inch;
l = length of air pipe, in feet;
d = diameter of air pipe, in inch .,
v = velocity, in feet per second.
R = ratio of compression relative to atmosphere.



Scale of time in seconds.

FIG. 28.

"Putting equation (22) in equation (20), there results

$$Q = V(1+n)\log_e\left[\frac{1+\frac{n}{R_o}}{1+\frac{n}{R_o}-\frac{P_o-p_1}{P_o}}\right]$$

$$= V(1+n)\log_e\left(\frac{1}{1-\frac{P_o-p_1}{P_o+np_o}}\right)$$

putting $\frac{P_o}{p_o}$ in place of R_o

"Now, as before stated, np_o will be quite small, as compared with P_o, and it can be neglected, if desired, to simplify the formulæ. Equation (22) would then become

$$Q = V(1+n)\log_e\frac{P_o}{p_1} \tag{23}$$

"This gives a simple formula for Q, the volume taken into the compressor while reducing the pressure from P_o to p_1 (in a tank full of air). To be precise, it should now be noticed that the operation begins properly with a pressure P_1 somewhat less than P_o. This is due to the expansion into the low-pressure pipes just after switching. This pressure P_1 can be found readily by the condition of the constancy of the sums of the products of the volumes by the pressures. Thus, equating the sums just before and after switching, there results

$$P_1(V+2n) = P_oV(1+n) + p_onV$$

or

$$P_1 = \frac{P_o(1+n)+np_o}{1+2n} \tag{24}$$

P_1, thus found, would be put in place of P_o in equation (23).

"The effect of friction in the air pipe between the tank and the compressor must now be considered.

"When the pressure of the intake of the compressor is P_x, that in the tank from which the air is drawn will be greater by the

amount lost in friction while passing through the pipe. The equation for this loss is, in form,

$$f = c\frac{l}{d}v^2R$$

where c is an experimental coefficient. From the best experimental data obtainable, it is found to be about 0.000002, when

f = lost pressure, in pounds per square inch;
l = length of pipe, in feet;
d = diameter of pipe, in inches;
v = velocity of air in pipe, in feet per second;
R = ratio of compression, in atmospheres.

In many rules for computing the loss by friction, the factor R is erroneously omitted. In this case $R = \frac{P_x}{14.7}$ and, therefore, is variable, but in any installation all are constant in the formula except P_x. Then, for simplicity, let

$$\frac{0.000002\, l}{14.7\, d}v^2 = k \tag{25}$$

"Then the lost pressure would be kP_x and, in equation (18), $P_x\,(1 + k)$ should be put in place of P_x, but this will in no way change the process by which equation (23) is derived. With this change equation (23) becomes

$$Q = V\,(1 + n) \times (1 + k) \log\frac{P_1}{p_1}$$

"If the compressor takes in a volume, q_a per second, the time consumed in working the pressure down from P_1 to p_1 is

$$t_1 = \frac{Q}{q_a} = \frac{V}{q_a}(1 + n)\,(1 + k) \log\frac{P_1}{p_1}$$

"During the remainder of the time in one cycle, the water is flowing into the tank, following up the air, and keeping it at nearly constant pressure (when the height of the tank is only a few feet); in other words, for every cubic foot of air taken out,

a cubic foot of water flows in. Hence, evidently, the time consumed in this last period of the cycle is

$$t_2 = \frac{V}{q_a}$$

and the total time,

$$T = t_1 + t_2 = \frac{V}{q_a} + \frac{V}{q_a}(1 + n)(1 + k)\log\frac{P_1}{p_1}$$

If q_w is the average rate of delivery of the water, evidently

$$q_w = \frac{V}{T}$$

whence

$$q_a = q_w\left[1 + (1 + n)(1 + k)\log\frac{P_1}{p_1}\right] \tag{26}$$

which is the desired equation.

"In practice k should not exceed 0.1, and will usually be less. If great precision is to be attempted, equation (26) must be solved by a tentative process, for k is a function of q_a. k may be first assumed as 0.1, to get an approximate value of q_a, whence v in tentative process, for k is a function of q_a. k may be first assumed as 0.1, to get an approximate value of q_a, whence v in equation (25), and a closer value of k. This will be sufficiently close for practice.

"It is probably useless to attempt extreme precision in these computations, on account of temperature changes which cannot be formulated. Hence, as a safe and simple working formula, the following may be used:

$$q_a = q_w\left[1 + 1.1(1 + n)\log\frac{P_o}{p_o}\right] \tag{26a}$$

p_o will commonly be near atmospheric pressure (or 15), that is, when the tanks are near the surface of the water, but it may be greater or less, according to whether the tanks are submerged or placed above the water. Inspection of equation (26) reveals the fact that the greater p_o is, the less will be q_a. For this reason there is an advantage in having the tanks submerged.

Evidently, if the air is heated by contact with hot surfaces while entering the compressor, the effective intake capacity is reduced. To allow for this circumstance, q_a, as above computed, should be multiplied by $\frac{\tau_2}{\tau_1}$, where τ_1 and τ_2 are the absolute temperatures before and after entering the compressor, respectively.

"**Maximum Rate of Work.** — The compressor capacity having been determined, the next problem in the design of a plant is to find the maximum rate of work for which provision must be made in the steam end of the compressor. The nature of this problem can best be presented by first studying the case of isothermal compression. In this the well-known formula for work, using the symbols heretofore applied, is

$$\text{Work per second} = P_x q_a \times \log \frac{P_o}{P_x} \tag{27}$$

"In this, P_x is variable, and, evidently, the work will be 0 when $P_x = 0$, and again, when $P_x = P_o$ (since $\log 1 = 0$), and, by the method of calculus, it is found to be a maximum when $\log \frac{P_o}{P_x} = 1$; that is, when $\frac{P_o}{P_x} = 2.72$.

"Note that hyperbolic logarithms must be used in all the foregoing equations as they appear. If common logarithms are to be used, multiply by 2.3.

"Inserting the condition for a maximum in equation (27) and reducing to foot pounds per second, there results

$$\text{Maximum work} = 52.9\, P_o q_a$$

"A curve showing the work by equation (27) is given in Fig. 25. In practice the curve does not reach zero at either end.

"To find the maximum work when temperature changes are considered, one must start with the established formula for work when compression is adiabatic, viz.:

$$\text{Work} = \frac{n}{n-1} P_x q_a \left[\left(\frac{P_o}{P_x} \right)^{\frac{n-1}{n}} - 1 \right] \tag{28}$$

where n is the 'temperature exponent' and equals 1.41 when no cooling occurs.

"By the methods of the calculus equation (28) will be found to be the maximum when $\left(\frac{P_o}{P_x}\right)^{\frac{n-1}{n}} = n$; or when $P_x = \frac{P_o}{n^{\frac{n}{n-1}}}$.

This, inserted in equation (28), gives

$$\text{Maximum work} = \frac{P_o q_a}{n^{\frac{1}{n-1}}} \tag{29}$$

When $n = 1.41$ maximum work $= 62.3\, P_o q_a$ foot pounds per second.
When $n = 1.25$ maximum work $= 59.0\, P_o q_a$ foot pounds per second.
When $n = 1.00$ maximum work $= 52.9\, P_o q_a$ foot pounds per second,

the last number having been derived by analysis of equation (27).

"As a simple approximate rule, the maximum horse-power rate may be taken as $0.1\, P_o q_a$.

"This maximum rate should not be confused with the average.

"**Efficiency.** — The only loss of energy chargeable to this system is that caused by the drop in pressure due to expansion into the low-pressure pipe just after switching. This drop is shown in equation (24). The ratio of this change of pressure is

$$\frac{P_o}{P_1} = \frac{1 + 2n}{1 + n + \frac{p_o}{P_o} n} = r$$

for simplicity. The necessary work to restore this pressure is

$$P_o V (1 + n) \log r$$

while the useful work done during a cycle is $(P_o - 14.7)\, V$, that is, the water displaced multiplied by the gauge pressure. Hence

$$\text{Efficiency} = E = \frac{(P_o - 14.7)\, V}{(P_o - 14.7)\, V + P_o V (1 + n) \log r}$$

$$= \frac{1}{1 + \frac{P_o}{P_o - 14.7}(1 + n) \log r} \tag{30}$$

Losses due to heat and friction are not included. It should be noticed that this loss is dependent on n. Its amount is illustrated by the following: E changes but little with other values of P_o and p_o.

$$\left.\begin{aligned} P_o &= 100 \\ p_o &= 14.7 \end{aligned}\right\} \begin{aligned} n &= 0.1 \quad 0.2 \quad 0.4 \quad 0.6 \quad 0.8 \quad 1.0 \\ E &= 0.91 \quad 0.85 \quad 0.74 \quad 0.66 \quad 0.60 \quad 0.55 \end{aligned}$$

"**Friction Losses.** — In the operation of a plant the velocity in the intake pipe will be constant but the pressure variable, while, in the discharge air pipe, the pressure will be constant and the velocity variable. According to equation (25), the loss in the intake is, in pounds per square inch,

$$\frac{0.000002}{14.7}\frac{l}{d}V^2P_x = kP_x = f_i \tag{31}$$

and the loss due to the same air passing through the discharge pipe at the pressure P_o is

$$\frac{0.000002}{14.7}\frac{l}{d}\left(\frac{P_x}{P_o}V\right)^2 P_o = k\frac{P_x^2}{P_o} = f_i\frac{1}{R_x} \tag{32}$$

To find the friction losses at intervals in the cycle, or to show such by a curve, assume convenient intervals of time (5 or 10 seconds) which indicate by t_x. Then

$$t = \frac{Q_x}{q_a} = \frac{v(1+n)\ (1+k)\log\frac{P_1}{P_x}}{q_a}$$

Whence, adapting to common logarithms,

$$\log_{10} P_x = \log_{10} P_1 - \frac{t_x}{\dfrac{V(1+n)(1+k)}{q_a}}(0.434) \tag{33}$$

Thus, tabulate P_x corresponding to t_x and apply the slide-rule to get the friction losses from equations (31) and (32).

At any time the rate of water discharge will be

$$w_x = \frac{P_x}{P_o}q_a$$

"This can be tabulated with the other quantities, and the friction loss in the water pipe worked out accordingly by well-known formulæ. Curves worked out by the foregoing methods are shown in Fig. 28."

Based on Prof. Harris' formula, Table 5 has been prepared by Mr. H. T. Abrams and included in a lecture delivered before the Junior Class of Columbia University.

TABLE 5

SIZE OF COMPRESSOR, PIPES, ETC.

For various heads based on 100 gallons of water per minute. The sizes for other quantities of water will be directly proportional.

Lift in feet	Capacity of compressor in cubic feet per minute. Piston displacement for 100 gallons per minute	Maximum I.H P. of air cylinder	Maximum I H.P. of steam cylinder	Average H.P. of steam cylinders	Area of air pipe in square inches for each 100 gallons. Capacity of plant	Area of water pipe in square inches for each 100 gallons. Capacity of plant
50	39 84	2 74	3 22	2 80	0 96	7 70
60	42 78	3 28	3 85	3 37	1 03	8 25
70	45 30	3 85	4.53	3 93	1 09	8.73
80	47 70	4 45	5 22	4 49	1.14	9 12
90	49 80	5 03	5 91	5.05	1.20	9.60
100	51 84	5 67	6 67	5 61	1 25	10 00
110	53 64	6 31	7.43	6.17	1 29	10.30
120	55 44	6 96	8 18	6.73	1 33	10 60
130	57 00	7 62	8 97	7.29	1 37	10.95
140	58 50	8 30	9.75	7 85	1.41	11.30
150	59 94	9 00	10 60	8.41	1 44	11.50
160	61.38	9 75	11.45	8 98	1 47	11 75
170	62.64	10.42	12 25	9 54	1 50	12.00
180	63 84	11 13	13 08	10 10	1.53	12.25
190	64 98	11 85	13 95	10.66	1 55	12.40
200	66.12	12.76	15 00	11.22	1 58	12.65
210	67 20	13.35	15.70	11.78	1 62	12.95
220	68.28	14 09	16 60	12 34	1.64	13.10
230	69 24	14 92	17.35	12 90	1.67	13.35
240	70.20	15 68	18.45	13 46	1.69	13.50
250	71.10	16.46	19 35	14 02	1.71	13.70
260	72.00	17.24	20 25	14 58	1.73	13.82
270	72.84	18.00	21.20	15.14	1.75	14.00
280	73.56	18.80	22.10	15.71	1.77	14.20
290	74.28	19 60	23.10	16.27	1.79	14.30
300	75.06	20.45	24.00	16.83	1.80	14.40

These tables assume that tanks are fully submerged.

Efficiency. — *The Ingersoll-Rand Co., manufacturers of the* return-air system, state in their catalogue, No. 75, that the efficiency averages about 55 per cent and, even under unfavorable conditions, the efficiency has never fallen below 40 per cent. This efficiency is computed by dividing the water horse power by the indicated horse power in the steam cylinder of the compressor, and, consequently, all losses of transmission, compression, etc., are included.

Performance of a Return-air System. — Like the plain displacement pumping system, the return-air system has been seldom carefully tested, and, consequently, comparatively little is known of the everyday performance of the system in-so-far as working economy and over-all efficiency are concerned. This is unfortunate because the field open to apparatus of this kind is almost unlimited and the efficiency and practical advantages would seem to entitle it to a broader exploitation.

Undoubtedly the most accurate and carefully conducted test of return-air system was made by Mr. Arthur H. Diamant, C.E., and published in Vol. LIV of the "Transactions of the American Society of Civil Engineers." The machinery was furnished by the Pneumatic Engineering Co., and installed in Shaft No. 25 of the Croton Aqueduct. Mr. Diamant's description of the plant, the difficulties attending its installation and the method of testing with the tabulated results are well worth repeating here. The following is Mr. Deamant's paper, and is entitled *The Installation of a Pneumatic Pumping Plant.*

THE INSTALLATION OF A PNEUMATIC PUMPING PLANT.

"Before proceeding with the description of the pumping plant, which is to be used in case of emergency only, the writer deems it advisable to give a brief statement as to the necessity for its installation.

"As is generally known, the City of New York receives its water

supply through the New Croton Aqueduct, which begins at the inlet gate-house, near the Old Croton Dam, Croton Lake, and, after reaching Shaft No. 24, on the Bronx side of Washington Bridge and near it, passes under the Harlem River to Shaft No. 25 and thence to the terminal gate-house at One Hundred and Thirty-fifth Street, near Amsterdam Avenue.

"Provision has been made for emptying the aqueduct whenever necessary. The inlet gates at Croton Lake can be closed, thus preventing water from entering the aqueduct. To empty that portion between Croton Lake and Washington Bridge, there are blow-off gates at Shaft No. 9, Pocantico; at Shaft No. 14, Ardsley; at Shaft No. 18, South Yonkers; and at Shaft No. 24, near Washington Bridge. There are also blow-off pipes on the stretch between Shaft No. 25 and the terminal gate-house so that the

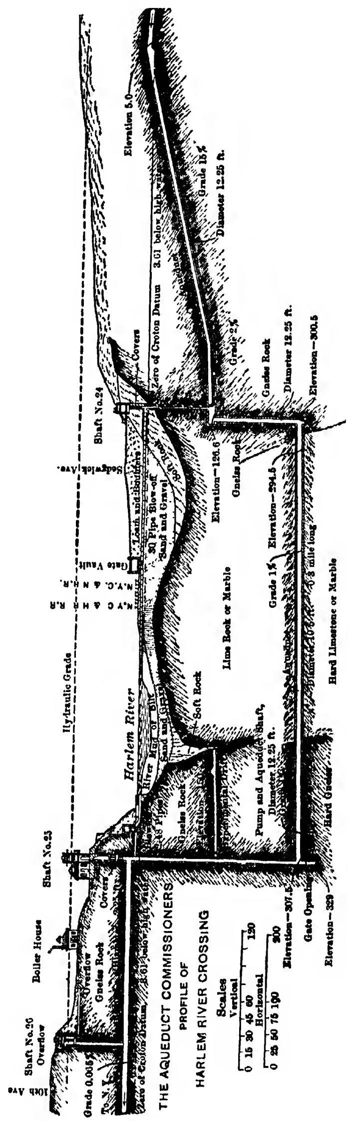


FIG. 29.

aqueduct can be made to empty itself, with the exception of that portion constituting the Harlem River crossing or siphon, Fig. 29. This siphon is emptied by Shaft No. 25, the pump-shaft.

"Shaft No. 25 (see Fig. 30) is really a double shaft, the northerly one being the aqueduct-shaft, and the southerly one the pump-shaft.

"The aqueduct-shaft is 12.25 feet in diameter, and, at a point about 10 feet above high water, the aqueduct continues on its way to the terminal gate-house. The pump-shaft, also 12.25 feet in diameter, is completely lined with iron, and contains a sump extending 21.75 feet below the bottom of the siphon tunnel. An opening, 1 foot 8 inches by 2 feet 6 inches, and 3 feet below the invert of the tunnel, regulated by a gate, admits the water into the pump-shaft. This gate, being 417 feet below the top of the shaft, is of composition metal, moving in solid composition grooves, and is designed so that no obstructions can accumulate in the frame. It is raised by a square stem, 3.5 by 3.5 inches, guided every 12 feet, and contained in a 3-foot pipe built in the masonry. This pipe also contains a ladder reaching from the top (Elevation 84.5) to the bottom (Elevation 312.75). A plan of the connection is shown in the 'Section through *AB*,' Plate I. As each shaft is under the hydraulic grade, it can be closed by a double set of manholes with covers. For the purpose of blowing off the water, each shaft is connected with a 48-inch cast-iron pipe, with two gates, and discharging into the river.

"Over the pump-shaft was erected a bucket-hoist, composed of two alternating buckets, each of 1390 gallons capacity. These were raised and lowered by a horizontal steam engine capable of emptying each in 0.5 minute. Fig. 31, prepared by Mr. F. S. Cook, Engineer in Charge of the Draughting Bureau of the Aqueduct Commissioners, shows the volumes of water to be lifted in emptying the siphon. With this plant, it would have taken from 15 to 18 hours to accomplish this task, provided the engines could have continued at the aforesaid rate. As shutting down the aqueduct would entail serious inconveniences, the

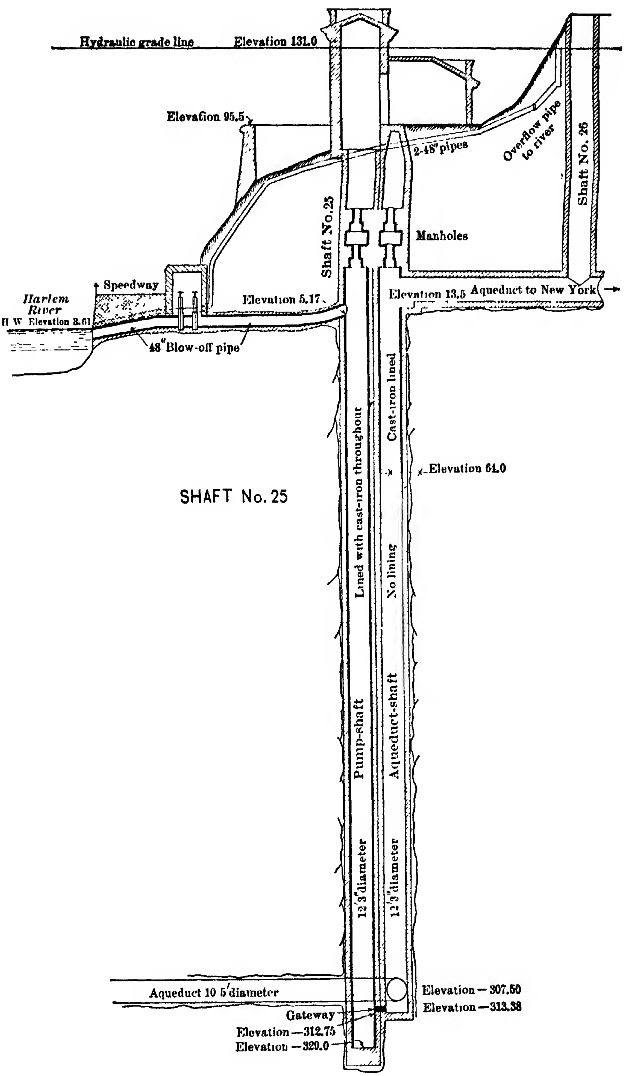

Hydraulic grade line
Elevation 131.0
Elevation 95.5
2-48" pipes
Overflow pipe to river
Shaft No. 26
Shaft No. 25
Manholes
Speedway
Harlem River
H W Elevation 3.61
Elevation 5.17
Elevation 13.5
Aqueduct to New York
48" Blow-off pipe
Lined with cast-iron throughout
Cast-iron lined
Elevation 61.0
SHAFT No. 25
No lining
Pump-shaft
Aqueduct-shaft
12'3" diameter
12'3" diameter
Aqueduct 10 5' diameter
Elevation — 307.50
Elevation — 313.38
Gateway
Elevation — 312.75
Elevation — 320.0

FIG. 30

present water consumption being about 296,000,000 gallons per day, the Aqueduct Commissioners deemed it necessary to install a pumping plant which could empty the siphon in 12 hours or less, as every hour gained would be of material advantage. Accordingly, bids were received for such a plant, and the contract was awarded to the Pneumatic Engineering Company, who pro-

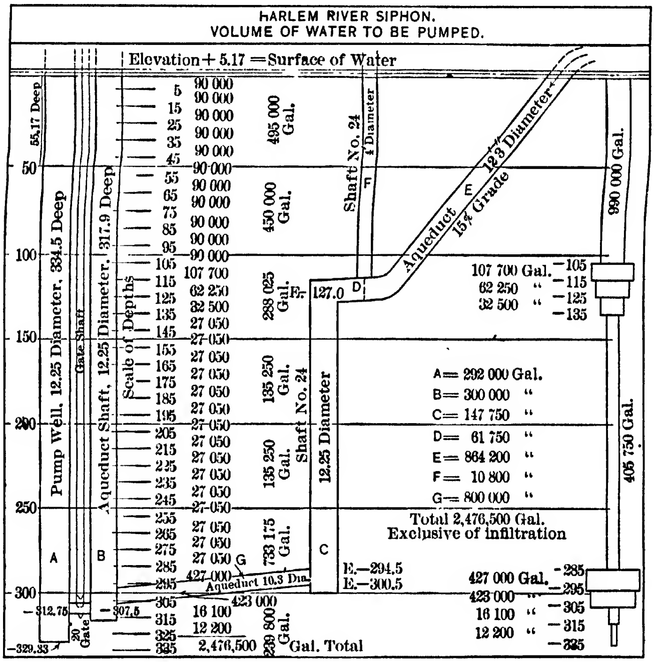


FIG. 31.

ceeded to install the Harris System of pneumatic pump. The contract specified that 2,500,000 gallons be raised 337 feet in 12 hours, with a bonus of $5,000 for every hour less than 12 hours, and a penalty for each hour longer.

"This system, briefly described, consists of a 27 by 48-inch Comstock compressor, twin-connected; the steam engine, 24 by

48-inch, being the improved horizontal type, with Corliss valves. Free air, being compressed, passes through coolers, through a switch apparatus, down pipes into four water tanks working in pairs at the bottom of the pump-shaft. An auxiliary compressor supplies the necessary air for running the plant with the greatest efficiency. (See Plate II.) The system is described more fully in the latter part of this paper.

"The installation was attended with peculiar difficulties. Leaks have developed in the pump-shaft, since its construction, keeping it full of water up to the blow-off pipe. Weir measurements taken in this pipe show a leakage of about 200 gallons per minute. As there was no way of emptying the shaft and keeping it empty, all work had to be done on an erecting platform built near the blow-off pipe. A 5-inch ejector kept the water about 15 feet below the platform.

"As before stated, the shaft is below the hydraulic grade. If the gate between the aqueduct-shaft and the pump-shaft were to be opened, and the blow-off gates closed, it would be necessary to put on the covers of the manholes in the diaphragms. For this reason, the air-pipes leading down to the four water tanks (see Plate I) could not pass through the manhole openings, but had to pass through four holes bored through the brickwork and iron lining of the two diaphragms. As seen in the drawing, these diaphragms are each about 9 feet thick, with a space of 6.7 feet between them. The Rand Drill Company's Davis Calyx drill was used in making the four holes, each 9 inches in diameter, steel shot being used for the cutting surface. Cores, from 3 to 4 feet long, were taken out, showing the efficiency of drills of this style.

"For the purpose of lining these holes, and making them continuous between diaphragms, a 6.75-inch cast-iron pipe, with a flange at one end, was placed in each hole of the upper diaphragm, the flange resting on its upper side. A similar pipe was placed in each hole of the lower diaphragm, the flange being bolted to the iron lining of the underside thereof. A short length of pipe, of the same inside diameter and with a hub on each end, con-

nected the two pipes of each hole. Perfectly tight joints were made with lead. The utmost care had to be taken in pouring the lead as there was a great deal of moisture in the space between the diaphragms.

"While these holes were being drilled, the old bucket-hoist engines were taken apart and removed, the buckets having been taken out of the shaft previously. The old brick foundations also had to be cut away to make place for the new ones. The new foundations, both for the compressor and the steam engine, are each 9 feet high, 8 feet wide and 32.5 feet long. They are composed of a 1: 3: 5 concrete mass finished with a 1-foot granite coping stone over the entire top. The foundation of the auxiliary compressor is also of concrete, with granite coping stones.

"The different parts of the system having arrived, the first pair of tanks was taken into the engine-house and placed in proper position near the top of the shaft at Elevation 84.5. The second pair was then placed also. These tanks are 17.5 feet high and have an inside diameter of 4 feet 2 inches. A cage, operated by a small Otis steam engine, carried the men from the top of the shaft down to the blow-off at Elevation 5.17. This cage could be shifted to pass through the north or south opening as necessity required. With the four tanks, a clearance for the elevator cage, a 36-inch water main and gate, and the lifting machinery for the connecting gate between the aqueduct- and pump-shafts, there was very little room to spare.

"Fig. 32 is a front view of the tanks and fittings assembled at Elevation 84.5, before being taken apart to be lowered to the erecting platform. The photograph shows one pair of tanks, the manner in which they are connected, the intake pipe with the 10-inch check-valves admitting water into the tanks, the 14-inch discharge pipes with 10-inch check-valves opening outward, the 5-inch air.pipes, and the I-beams and hangers for lowering the tanks. The discharge pipe passes into each tank within about 6 inches of the bottom, a cone at this point guiding the water from the tank into the discharge pipe (see Plate I). Near the top of the inlet pipe is a cast-iron groove which is to slide

along the old bucket-guides in the shaft. Grooves similar to this are on the plates connecting the tanks near the bottom, and also on the plates in the back of the tanks. The I-beams and hangers to which the wire cables of the lowering apparatus are attached can be seen near the extreme top of the photograph.

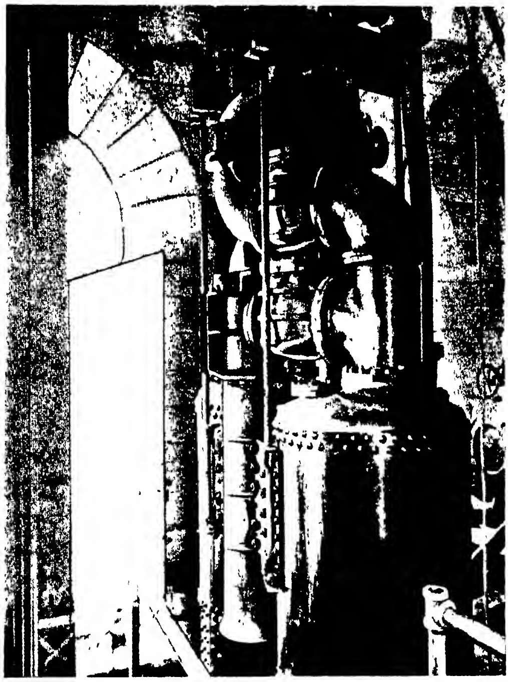


FIG. 32.

"On top of the Y, connecting the discharge pipes of the two tanks, a T was placed, to which, by means of an elbow, an air chamber was fastened to prevent water ram. A $\frac{3}{8}$-inch pipe, with a check-valve, leads from this elbow to the top of the shaft, being used to charge the air chamber. At this point, also, the discharge pipe and the two air-pipes were fitted with swivel

joints, so that, even if the tanks did not rest perfectly level on the bottom, the pipes could be carried up vertically, by means of these joints, which were perfectly tight.

"The 14-inch discharge pipes are rolled-steel tubes with cast-steel flanges. These were shrunk on the tubes, and the ends of the latter were upset. The pipes were delivered in this condition, but, as the upset ends projected from $\frac{1}{8}$ to $\frac{1}{2}$ inch beyond the faces of the flanges, this part had to be removed, otherwise no tight joints could have been made. To return the pipes to the foundry would have caused the loss of too much time, therefore a lathe was rigged up outside of the engine-room. Sand was strewn over the space to be used, some iron-grating floor-plates from the shaft-house were embedded in the sand, and several pieces of the coping stone of the old engine foundation were placed on top of the plates. Two 15-inch I-beams, 20 feet long, with a space of about 4 inches between them, were laid on top of the stones and fastened firmly by steel rods passing down to the grating plates. The I-beams were leveled carefully, and the 14-inch pipes, being laid on top, were thus also level. The pipes were held fast by a V-shaped clamp at each end. A chuck holding the cutting tool was geared to a shaft which was revolved by being belted to a small vertical steam engine. The tool was fed automatically into the flange to be faced by means of a star wheel which, at each revolution of the chuck, would strike one of its prongs against a projecting board, thus causing the tool to cut deeper. This apparatus proved to be very efficient, as the faces of the flanges were made absolutely at right angles to the axis of the pipe, thus ensuring a perfectly straight column when the lengths were bolted together. There were thirty-two pieces to be faced on each end, and the entire work was completed in 11 days.

"While this was being done, men were engaged in placing an erecting platform, just below the blow-off. This consisted of brackets fastened to the east and west sides of the shafts with bolts let into the iron lining. On these brackets was placed a 15-inch I-beam, which was bolted down. Resting upon this

beam, and also upon brackets fastened to the north and south sides of the shaft, were placed 12-inch timbers, over which a 3-inch plank flooring was fastened. All drilling for stud-bolts was done with a pneumatic drill, the air being supplied by a small Rand Drill Company's compressor on top of the shaft.

"Just before the erecting platform was completed, there occurred the only accident during the entire installation. Fortunately, this was attended with no serious results. In order to lower the tanks, it was necessary to remove the catch-basin into which the buckets of the old system discharged. The bottom plate is shaped like a segment of a circle, with a chord of 11 feet 6 inches, and a rise of 3 feet 10 inches, the radius of the arc being 6 feet $1\frac{1}{2}$ inches. The weight of the plate was about 2000 pounds. Two 8 by 12-inch holes had been cut into it some years ago. Two workmen were on top of the plate passing a chain through one of the holes, when it slipped from its bearings, tipped over, and dropped to the bottom, a distance of 333 feet. The men were thrown into the water, but, with the exception of some bruises, were not injured seriously.

"Before lowering the tanks, this plate had to be recovered. Accordingly, the writer, with an assistant, sounded every foot of the bottom of the shaft with a steel tape and lead weight, and was fortunate enough to locate one of the 8 by 12-inch holes. A chain with a hook at the end was fastened to the elevator cable, lowered to the bottom, and guided from the erecting platform by a ribbon tape. The hole, located by previous soundings, was again found, and, after two or three trials, the plate was hoisted to the surface. A small corner broken off was the only damage the plate sustained.

"As the soundings indicated some silt at the bottom of the shaft, sixty bags of sand were dumped into the shaft, the bottom being thus fairly leveled.

"The first pair of tanks and fittings, which had been assembled at Elevation 84.5, was now taken apart, preparatory to being lowered to the erecting platform. The elevator cage was hoisted out of the shaft, so that its cable could be used. Each tank was

first lowered to the top of the first diaphragm with a block and fall. The elevator cable was then attached and the tank was lowered to the erecting platform. The tank was hung with such exactness that it passed through the manhole without binding, although there was only about $\frac{1}{4}$-inch clearance. The **T**'s and **Y**'s, and other parts having been lowered, the first pair of tanks was again assembled, and placed in the exact position in which they would have to be lowered to the bottom.

"The first pair of tanks being out of the way, the hydraulic lowering apparatus was set up in the south half of the shaft at Elevation 84.5 (see Plate I). This apparatus consists of a cylinder with a plunger having a stroke of about 21 feet, a balanced elevator-valve and pressure pump, two **A**-frames on top of the upper diaphragm and two on the erecting platform. In the timber bents, 10 by 10-inch beams were used. Plow-steel wire cables were used, and were fitted with sockets at each end. Their breaking strain was 98 tons. A 60-foot length of cable reached from the lifting **I**-beams of the plunger to the holding **I**-beams above the tanks, the plunger being about 1 foot from the top of its stroke. Two cables were used for each pair of tanks. The plunger was now raised as high as it could go, the tanks thus being raised about 1 foot, and the planks and beams on which they had been resting were removed. The water in the hydraulic cylinder having been allowed to exhaust, the tanks were lowered 20 feet. The **A**-frame on the upper diaphragm had been placed in position, and the sockets of the 60-foot cables rested on clamps which were now bolted on. These clamps in turn rested on **I**-beams on top of the bents. While the whole weight rested on these **A**-frames, the pins, which held the sockets of the cables to the lowering **I**-beams of the plunger, were removed, the plunger was again raised to near the top of its stroke, and the longer **A**-frames on the erecting platform were placed in position. A 20-foot length was added to each cable, a length of 14-inch pipe to the discharge pipe, a length of 5-inch pipe to each of the air-pipes, and a length of $\frac{3}{8}$-inch pipe to each of the charging pipes of the air chambers. Each joint was tested under an air pressure

of 150 pounds, as were also the tanks and valves before lowering, to insure perfect tightness of all joints. The load was now raised slightly, the clamps removed, and the tanks lowered 20 feet. The procedure being the same, the tanks were lowered another 20 feet, this time, however, the clamps rested on the I-beams of the lower A-frames. The three 20-foot lengths of cable were now taken out and replaced by a 60-foot length, and the cycle again started. In this manner the first pair of tanks was safely lowered to the bottom of the shaft, a depth of 332 feet. Pipe-guides or stays were fastened to the pipes every 60 feet, the ends of the stays sliding along the old bucket-guides. The total weight lowered was estimated at about 40 tons. The elevator cage was now shifted to the other side of the shaft, as was also the hydraulic lift, and the second pair of tanks was lowered in the same manner as the first. All parts before going down were painted both outside and inside with two coats of 'Nobrac' paint.

"The top of the discharge pipe of the second pair of tanks was about 4 inches above that of the first pair. Short lengths of discharge pipe added to each brought them to the same level. To this discharge pipe was added a T-piece. A Y connected the T-pieces, and a 20-inch goose-neck of galvanized-iron pipe was bolted to the Y. This pipe discharged into a catch-basin at the entrance of the blow-off, the bottom plate being the same one that fell to the bottom of the shaft, the sides being smaller than those of the old one. Cover-plates were bolted to the top of the two T-pieces. The four 5-inch air-pipes were now carried up to Elevation 84.5. Glands, through which these pipes passed, were bolted to the flanges of the iron lining of the holes through the upper diaphragm, so that, if the covers of the manholes were put on, the water could not pass between the air-pipes and the lining of the holes. When the shaft was built, a 4-inch pipe from the bottom of the lower diaphragm to a point 1 or 2 feet above the hydraulic grade served as an air-vent when the manhole covers were on. This pipe had been removed, above the upper diaphragm, and the two $\frac{3}{8}$-inch pipes were carried through this 4-inch opening to the top. This 4-inch pipe was afterward replaced.

"At Elevation 84.5 the four 5-inch air-pipes were connected with two manifolds, and from each of the manifolds an 8-inch air-pipe led to the switch. By means of these manifolds, any two tanks could be cut out of service and the pumping done with the other two. (See elevation of general plan, on Plate II.) The two $\frac{3}{8}$-inch pipes from the air chambers were connected by a **Y**, and led to the 8-inch air-pipe from the after-coolers to the switch. A $\frac{3}{8}$-inch pipe from the same point, with a pin-valve to allow air to leak into it, was hung down into the shaft, within 6 feet of the bottom, passing through the 4-inch opening through the diaphragms. This shows the pumping level and the pressure due to the head of water in the shaft. An 8-inch pipe carries the return air from the switch to the compressor on the side opposite the free-air valve.

"The switch consists of a plunger, with a stroke of $6\frac{5}{8}$ inches, operated by a piston moving in a smaller cylinder. The air is introduced into this smaller cylinder by a valve which depends for its action on a piston in a small cylinder, which, in turn, is caused to move by the action of a disc-valve. (See Fig. 33.) The disc or diaphragm is 6 inches in diameter, with a movement of $\frac{3}{32}$ inch, and consists of two thin sheets of bronze and one sheet of steel. A $\frac{3}{8}$-inch pipe conveys the return air from a point near the top of the cylinder of the plunger to one side of the disc-valve and the $\frac{3}{8}$-inch pipe, which shows the pumping level and pressure due to the head of water, leads to the other side of this valve. This latter $\frac{3}{8}$-inch pipe, connected with the 8-inch pipe from the after-coolers to the switch, receives air through a pin-valve, and is also piped to a gauge on the gauge-board, so that the pumping level and the pressure due to the head of the water can be seen at a glance. A small reservoir on this line gives a constant supply of air. The small, return air-pipe is also piped to a gauge, showing the return pressures. The difference between the pressure due to the head of the water in the shaft, which for the same levels is constant, and the return pressure (which is varying constantly, and drops to zero when the switch acts), causes the disc-valve to move. As the operation of the switch requires

PLATE I.

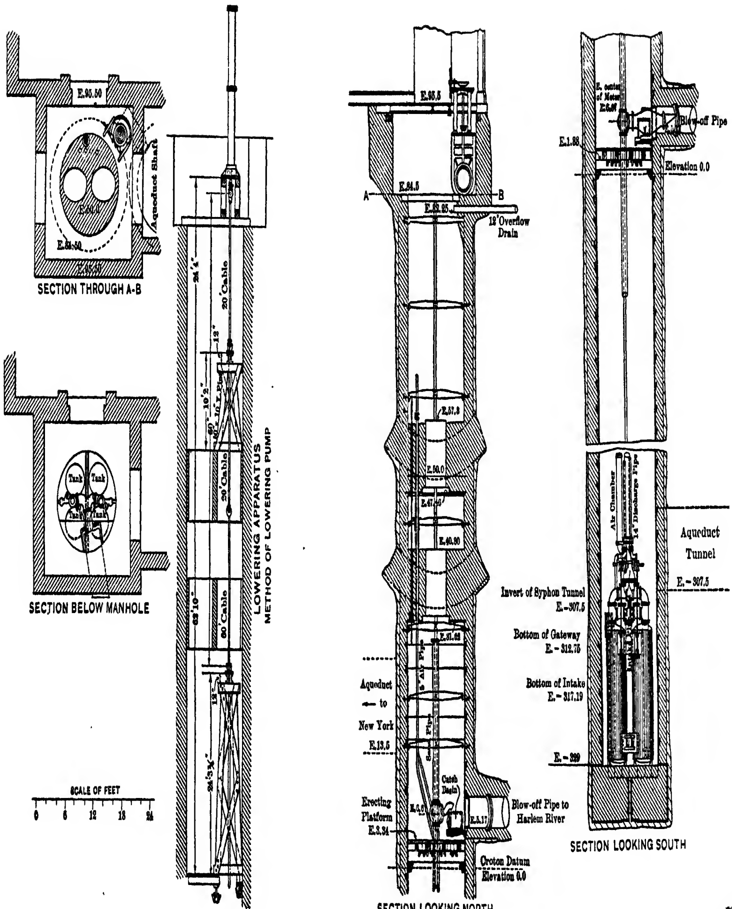
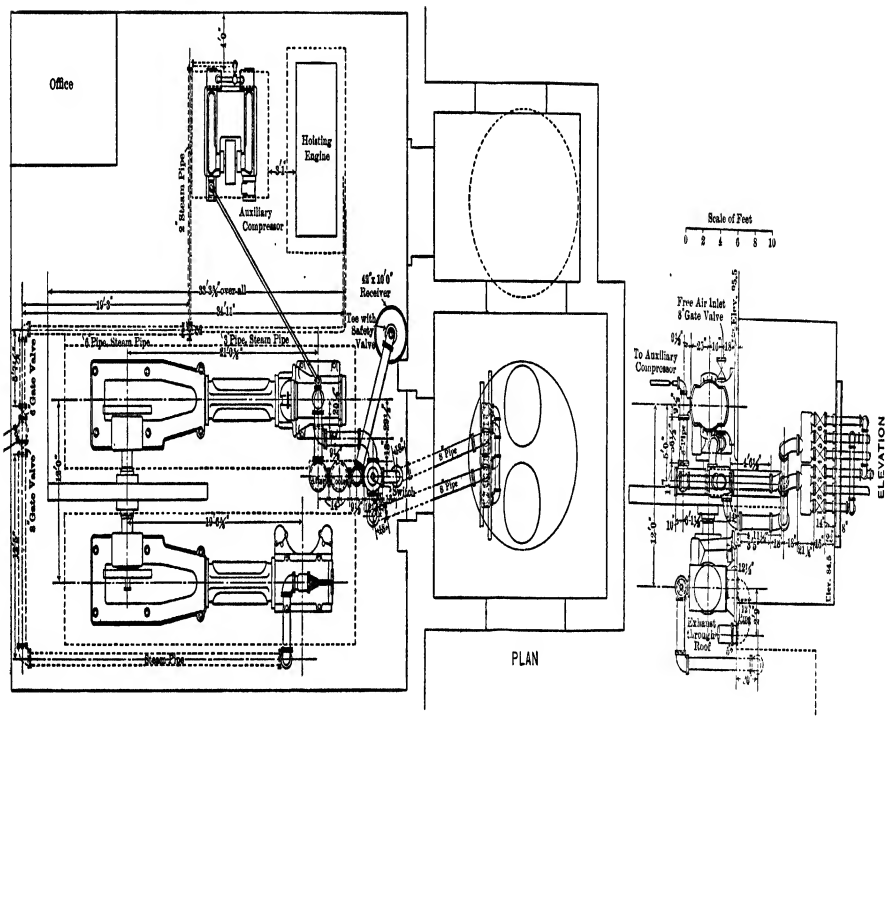

E.95.50
Aqueduct Shaft
SECTION THROUGH A-B
Tank
Tank
SECTION BELOW MANHOLE
SCALE OF FEET
0
6
12
18
24
24'4"
20'Cable
13"
10'2"
20'Cable
63'10"
60'Cable
12"
24'3¾"
LOWERING APPARATUS
METHOD OF LOWERING PUMP
E.95.5
E.64.5
A
B
12"Overflow
Drain
E.57.8
E.50.0
E.40.90
8"Air Pipe
Aqueduct
to
New York
E.13.5
Catch
Basin
Erecting
Platform
E.3.34
Blow-off Pipe to
Harlem River
Croton Datum
Elevation 0.0
SECTION LOOKING NORTH
Blow-off Pipe
E.1.88
Elevation 0.0
Air Chamber
14" Discharge Pipe
Aqueduct
Tunnel
E.-307.5
Invert of Syphon Tunnel
E.-307.5
Bottom of Gateway
E.-312.75
Bottom of Intake
E.-317.19
E.-339
SECTION LOOKING SOUTH

PLATE II.



Office
Hoisting Engine
Auxiliary Compressor
2" Steam Pipe
Receiver
Tee with Safety Valve
Steam Pipe
Gate Valve
Switch
8 Pipe
PLAN
Scale of Feet
0 2 4 6 8 10
Free Air Inlet
8" Gate Valve
To Auxiliary Compressor
Exhaust through Roof
ELEVATION

an air pressure of only about 50 to 60 pounds per square inch, a $\frac{3}{8}$-inch pipe from the 8-inch compressed-air pipe conveys the air through a reducing valve to the cylinder on top of the plunger, and to the piston of the small cylinder operated by the disc-valve. A reservoir on this line also ensures constant pressure. This

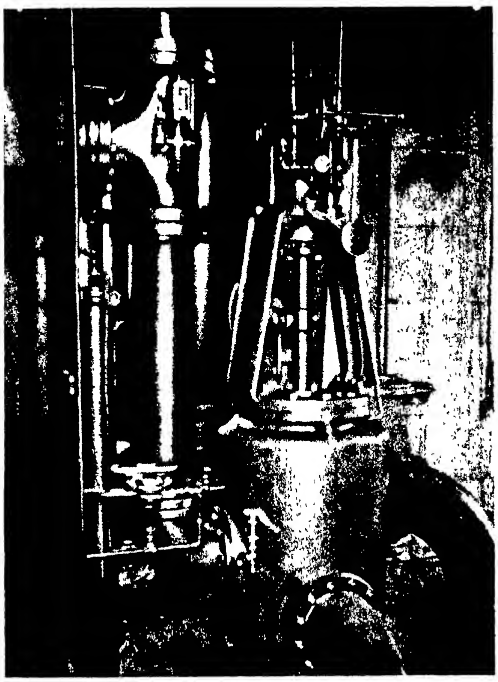


FIG. 33.

$\frac{3}{8}$-inch pipe also leads to a dial on the gauge-board, showing the pressures required to operate the switch. The disc-valve moving, due to the difference between the pressure caused by the head of the water in the shaft and the return pressure, allows air to enter the small cylinder above it, the piston moves, the valve controlled by this piston motion allows air to enter above or

below the piston in the cylinder above the plunger, the plunger acts and the air is sent alternately from one 8-inch air-pipe into the other, one of these 8-inch pipes always serving to return the air through the switch to the compressor. (See Plate II.) Provision is also made for operating the actuating valve by hand.

"The auxiliary compressor was set up, the large compressor and engine were adjusted, the piping was completed between the auxiliary compressor, the large compressor, the switch after-coolers, and the receiver, and the plant was ready for operation. Before pumping, all joints were tested as to tightness.

"The action of the plant is as follows: The large compressor is first started; the exhaust valve being closed, it requires about 312 revolutions of the fly-wheel to charge the system, the free air being compressed to 150 pounds per square inch. The free-air valve is now closed and the switch thrown over by hand. The compressed air passes from the compressor through the two after-coolers, into a receiver supplied with a safety valve, and also through the switch through one of the 8-inch pipes, through its manifold into the 5-inch air-pipes and into one pair of tanks. The air entering this pair of tanks forces the water through the discharge pipes and empties the tanks. As soon as this occurs, the return pressure from the other pair of tanks being less than the pressure due to the head of the water in the shaft, the actuating valve of the switch acts, the plunger moves, compressed air enters these tanks, while the air from the other pair is returning through the switch into the compressor to be used over again, and so on. A cycle consists of the number of revolutions of the fly-wheel necessary for the compressor to empty one pair of tanks to the point of starting to empty the other pair. The number of revolutions per cycle varies for different pumping levels, but is constant for the same level. If there are too many revolutions in charging the machine, or if there are too many revolutions per cycle, the air follows the water through the discharge pipes, and thus the system loses the air, necessitating the opening of the free-air valve of the compressor and recharging. After the plant has been working, a certain amount of air is lost,

and in order to keep up the proper number of revolutions per cycle the auxiliary compressor is started, and furnishes the air to keep the system working efficiently.

"Before the final test, many trials were run. Indicator diagrams of the Corliss engine were taken, and adjustments were made. The goose-neck discharge pipe was removed, and a 20-inch Gem meter placed in its stead. The cover-plates were taken off the T's, and a 20-foot length of 14-inch galvanized-iron pipe was added to each, so that no air would pass through the meter, but would escape through these pipes. (See Plate II.) Many over charges took place before the proper number of revolutions per cycle for different pumping levels was determined.

"The 20-inch Gem meter consists of a system of helicoids formed around a vertical central hub, revolving in a cylinder slightly greater in length, and having a diameter just large enough to receive it. A screen at the lower end of this cylinder serves to keep large objects from entering the meter. The axle of the hub is geared to the meter register, which contains six figures and reads thousands of gallons.

"As it was necessary to test the accuracy of this meter, before using it, to determine the efficiency of the pneumatic pumping plant, F. W. Watkins, M. Am. Soc. C. E., Division Engineer of the Aqueduct Commissioner's Engineering Department, assisted by the writer, made a test of the meter at the testing plant of the National Meter Company, in South Brooklyn.

"The test consisted of a comparison of the meter register records with weir measurements of the same volume of water. The water to be measured was elevated by a centrifugal pump operated by a Nash gas engine to a height which gave a head sufficient to force the desired amount through the meter. The water passed from the pump, through a screen, into a small forebay, thence through the meter into the L-shaped weir chamber. The base of the L is about 8 feet long and 8 feet wide, and the long side, constituting the main weir chamber, is 33 feet long, 12 feet wide and 6 feet deep below the level of the weir crest. Baffle boards, placed in the angle of the L, serve to break up any eddies

which may form. The water flowing over the weir drops into the pump-well, and the cycle is again started. (See Fig. 34. Figs. 34 and 35 were furnished by John H. Norris, M. Am. Soc.

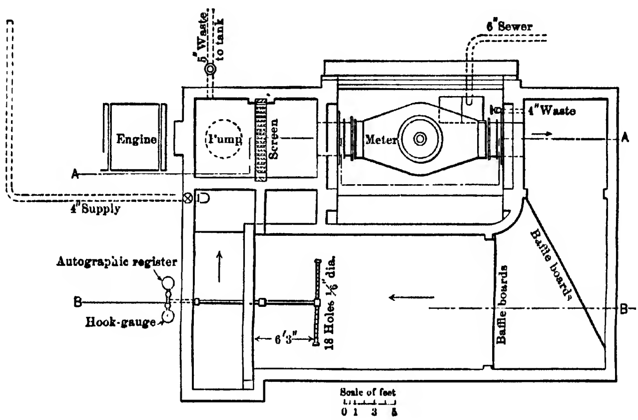


FIG. 34.

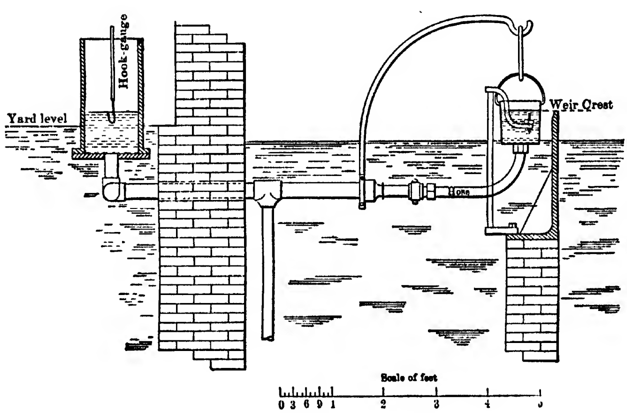


FIG. 35.

M. E., Assistant Engineer, National Meter Company, whom the writer takes this opportunity of thanking for his courtesies during the test.)

"The weir notch is of cast-iron plates, the plates forming the sides of the notch being adjustable, so that any length of wier, up to 8 feet, can be obtained. The crest was formed by beveling the down-stream face at an angle of 45 degrees, leaving a truly planed edge $\frac{1}{4}$ inch thick, the vertical sides having a similar bevel. The distance from the bottom of the weir chamber to the crest is 6 feet.

"The apparatus for measuring the head on the weir consists of two 12-inch cast-iron pipes set on end just outside of the catch-basin, one containing a float for the autographic record, the other the movable hook-gauge.

"These pipes are connected by a 2-inch pipe from which a 2-inch pipe leads through the wall of the catch-basin, with a valve at the other end. Another 2-inch pipe runs from this pipe to the bottom of the catch-basin, makes a right-angle bend, thence, parallel to and about 6 inches above the floor, it runs into the weir chamber and connects with a 2-inch pipe at right angles to it, parallel to the weir crest, and about 6.25 feet from the weir plate. This latter pipe was perforated with eighteen holes, each $\frac{3}{16}$ inch in diameter. (See Fig. 34.)

"Before starting the test, the relation of the hook-gauge and the autographic float-gauge to the weir crest was determined as follows: A fixed hook-gauge was fastened a few inches in front of the weir, and, by a spirit level, its point was adjusted exactly to the elevation of the weir crest. (See Fig. 35.) A bucket, with a rubber hose attached to the bottom, was hung over this fixed hook-gauge, the other end of the hose being attached to the 2-inch pipe leading to the movable hook-gauge, and to the autographic record. Water was poured into the bucket until the surface just covered the point of the fixed hook, when the water rose to the same elevation in the two 12-inch cast-iron pipes. The zero of the movable hook-gauge and the fixed pencil of the autographic gauge were now adjusted to correspond. The autographic gauge consists of a zinc float carrying a brass rod, to which a pencil is attached. Its point presses against a paper wrapped around a wooden cylinder revolving once an hour by

clockwork. Another pencil, attached to the frame holding the drum, marks a line corresponding to the elevation of the weir crest, so that the actual heads of water flowing over the weir can be seen at a glance.

"These preliminaries being over, the bucket was removed and the test begun. The weir opening was measured by a standard steel rule and was 4.2475 feet. Sufficient water from the city main was allowed to run into the catch-basin, the pump was started, and the water began to circulate. In order that the wind might not affect the test, the weir chamber was covered with boards.

"The Francis formula, with Hamilton Smith's correction, in the form of

$$Q = 3.29\left(L - \frac{H}{10}\right)H^{\frac{3}{2}}, \tag{34}$$

was used to calculate the quantity of water passing over the weir. In this formula

Q represents cubic feet of water per second;
L represents the length of the weir, in feet;
H represents the head, in feet.

"The velocity of approach was so small that it did not enter the calculation at all. The heads scaled from the autographic record checked very closely with the hook-gauge record. Table 6 is a summary of the tests, and is taken from the report of Major Watkins to William R. Hill, M. Am. Soc. C. E., then Chief Engineer of the Aqueduct Commission.

TABLE 6. — SUMMARY OF METER TESTS

Test	Head, in feet	Gallons per minute		Percentage meter to weir
		Meter	Weir	
First	0 296	1 002	1.003	99.90
Second	0.491	2.140	2.133	100.33
Third	0.603	2.905	2.895	100.34

"These tests proved the meter to be very accurate and consistent for different heads, and it was recommended by Major

Watkins as the standard measure for the pneumatic pumping plant at Shaft No. 25.

"As it was impossible at that time to shut down the aqueduct, so that the siphon could be actually emptied, it was decided to pump at different water levels in the pump-shaft. The water was first pumped out through the blow-off pipe, until its surface was about 50 feet below it, when the gate was closed far enough to allow only the leakage into the shaft to pass through, the remaining water running back into the shaft. After pumping at this level for an hour, the gate was opened and the water pumped down to 125 feet below the blow-off, when the gate was closed down again to allow only the leakage to run off. In like manner the plant was tested for levels 175,225 and 300 feet below the blow-off.

"The average volumes pumped per minute, as indicated by the Gem meter, were as follows:

At 88 feet below the blow-off....6290 gallons per minute.
At 125 feet below the blow-off....6020 gallons per minute.
At 175 feet below the blow-off....5220 gallons per minute.
At 230 feet below the blow-off....4286 gallons per minute.
At 298 feet below the blow-off....2180 gallons per minute.

Tables 7, 8 and 9 show the details of these tests.

"It had also been agreed to run an endurance test of 12 hours, pumping at a level about 175 feet below the blow-off, but, owing to the dismantling of several boilers, sufficient steam could not be obtained and the test was postponed for several weeks. In the meantime, the machinery was overhauled; a revolution counter was placed on the auxiliary compressor, and a small pump lubricator attached to the switch-plunger cylinder. A small steam pump was also connected with the line of water pipe leading from the 36-inch pipe to the water jackets on the large and auxiliary compressors and also to the after-coolers, as previous to this there was not sufficient water to keep the air properly cooled."

TABLE 7
(Jan. 27, 1904)

Time	Steam pressure	Mean air pressure	Mean switch pressure	Return pressure	Depth: Index gauge (*A*)	Depth: Tape and float (*B*)	Depth: Sum *A*+*B*	Depth: Assumed by gauge	Gauge head over meter	Total lift	Main compressors: Rev. per cycle	Main compressors: R. p. m.	Gallons pumped: In 5 minutes	Gallons pumped: In 1 minute
10–37	...	...	...	...	...	...	...	(31°)	...	...	...	...	...	...
10–43½	...	...	...	...	303	...	...	...	...	...	...	...	...	...
10–45	137	150	50	90	292	.	...	8	6.5	...	...	...	...	5000
10–50	135	155	50	100	248	49.9	297.9	62	6.0	...	...	47	40,000	8000
10–55	135	150	52	92	225	82.2	307.2	85	5.0	...	21	52	37,000	7400
11–00	135	148	52	80	206	110.2	316.2	104	5.0	...	26	56	34,000	6800
11–05	134	148	50	90	180	135.8	315.8	130	5.0	...	28	59	32,000	6400
11–10	135	147	50	90	156	160.7	316.7	154	5.0	...	30	61	31,000	6200
11–15	134	147	48	90	136	183.2	319.2	174	3.5	...	33	64	28,000	5600
11–20	136	143	48	90	118	204.4	322.4	192	2.5	...	38	63	26,000	5200
11–25	135	144	48	90	80	219.3	299.3	230	4.0	...	...	62	24,000	4800
11–27	Blow-out		...	...	72	...	...	238	...	...	50	64	12,000	2400
11–29	Blow-out		...	...	65	...	...	245	...	...	...	...	...	...
11–30	...	...	...	...	...	...	...	...	...	...	...	52	...	...

TABLE 7. — (Continued)

	315 revolutions to charge													
1–45	138	.....	.....	....	.....	... .	. ..	. ...	.....	.....	.....	.....	......	......
1–49¾	.....	Commenced to charge			.....	... .	.....	. .	... .	.....	.....	.....	......	......
1–55	145	138	.....	.....	.....	.....	... .	.....	.....	.....	.....	.....	......	......
			Charged											
1–55½	140	150	47	95	243	.....	.. ..	67	....	. ...	.....	.....	......	......
2–00	135	150	46	90	180	95.0	275.0	130	2.0	.. ..	23	...	28,000	5600
2–05	137	150	45	95	122	124.6	246.6	188	5.0	.....	22	56	33,000	6600
2–10	135	145	47	95	113	150.8	263.8	197	5.0	. ...	25	60	32,000	6400
2–15	135	145	48	90	92	173.0	265.0	218	5.0	. . .	29	61	28,000	5600
2–20	136	145	48	95	82	191.7	273.7	228	3.0	.....	35	62	26,000	5200
2–25	135	145	47	90	82	206.2	288.2	228	3.0	... .	43	63	20,000	4000
2–30	135	145	48	90	52	219.7	271.5	258	1.0	.. .	48	63	20,000	4000
2–35	135	140	47	90	90	229.7	319.7	220	1.0	. ..	48	62	17,000	3400
2–40	135	140	48	90	80	241.0	321.0	230	1.0	. .	. ..	62	16,000	3200
2–45	135	140	48	90	71	249.5	320.5	239	0.0	. ..	58	62	16,000	3200
2–50	135	140	48	90	63	258.1	321.1	247	. .	. . .	64	61	14,000	2800
2–55	136	138	49	90	55	264.9	319.9	255	. .	.....	70	62	13,000	2600
2–56½	.....	.....	.. ..	... .	.....	.. .	. ..	. .	... .	.. .	.. ..	.....	8,000	1600
2–57½	.....	. ...	... .	. ...	52	.. .	.. .	258	... .	.....	7	.....	......	......
2–58	.....	.....	. ...	. ...	... .	275.0	.....	.....	... .	.....	.....	161	......	......

Test stopped. Nails found in air valves of compressor.

TABLE 8

Tests of Pneumatic Pumping Plant, Croton Aqueduct
(Feb. 22, 1904)

Time	Steam pressure	Mean air pressure	Mean switch pressure	Return pressure	Depth: Index gauge (A)	Depth: Tape and float (B)	Depth: Sum A+B	Depth: Assumed by gauge	Gauge head over meter	Total lift	Main compressor: Rev. per cycle	Main compressor: R. p. m.	Gallons pumped: In 5 minutes	Gallons pumped: In 1 minute
10–09	312	Revolutions		.....	.....	.....	.....	.....	.....	..	.....	.....	.....	.....
10–15	Started	to pump	.....	.....	.....	.....	.....	.....	.....	..	.....	.....	.....	.....
10–15	135	140	60	.....	312	.....	.....	.....	8		14	59	.....	.....
10–20	135	155	55	80	285	.....	.....	.....	7	.	.....	57	.....	.....
10–22½	.....	.....	.....	80	263	50	313.0	.....	.....	.	16	.....	.....	.....
10–25	135	155	53	.....	253	63	319.0	↑	6	..	17	56	.....	.....
10–30	135	155	54	70	242	.....	↑		5	..	17	56	.....	.....
10–35	135	150	55	70	235	.....			5	..	17	56	.....	.....
10–40	134	150	50	75	233	.....			5	.	17	56	31,000	6200
10–45	135	150	50	65	232	.....			5	..	16	56	31,000	6200
10–50	135	150	55	70	231	.....	Average=320 0	320−233=87 o average depth	5	..	18	56	32,000	6400
10–55	135	153	54	65	230	.....			5	..	19	56	32,000	6400
11–00	135	150	58	60	230	.....			5	..	.....	56	31,000	6200
11–05	134	145	60	65	230	.....			5	.....	18	55	31,000	6200
11–10	134	145	60	60	231	.....			5	.....	18	56	31,000	6200
11–15	135	150	58	65	232	.....			5	.....	18	57	32,000	6400
11–20	135	150	59	60	233	.....			5	.....	18	56	32,000	6400
11–25	135	150	60	65	234	.....			5	.....	17	56	32,000	6400
11–30	132	150	60	65	235	.....			5	.....	17	56	31,000	6200
11–35	125	145	.....	.....	236	.....	↓	↓	5	.....	.....	56	33,000	6600
11–37½	.....	.....	.....	.....	237	83	320.0	.....	Dam in	.....	.....	.....	.....	.....
11–40	.....	.....	.....	.....	.....	.....	.....	.....	5	.....	.....	56	32,000	6400
11–41	.....	.....	.....	.....	222	97.8	319.8	.....	.....	.....	.....	.....	.....	.....
11–42½	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11–43½	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....

TABLE 8. — (Continued)

11-45	122	.....		....	205	115	320.0	.....	4 Dam in	.	.....	56	31,000	6200
11-47	.....	.. .	.	. .	194	125	319.0	. .	.....	.....	.....	.....	.....	.....
11-50	123	145	60	60	188	.	↑	↑	3	..	23	56	29,000	5800
11-55	128	145	57	60	189	.. ..			4	.....	. ...	56	29,000	5800
12-00	128	150	58	60	190	. ..			4	....	23	56	29,000	5800
12-05	130	145	60	60	192	.. .			4	....	.. ..	56	29,000	5800
12-10	130	145	60	55	192	.. .	Average=320	320−194=126 average depth	4	....	22	61	31,000	6200
12-15	130	145	58	55	194	..			4	....	21	61	28,000	5600
12-20	130	145	58	55	195	. ..			4	.....	21	61	31,000	6200
12-25	130	145	58	55	196	.. .			4	.....	22	61	30,000	6000
12-30	130	145	59	55	197	..			....	. ..	22	61	31,000	6200
12-35	128	147	59	55	197	. .			....	....	22	62	31,000	6200
12-40	128	148	60	60	197	. ..			....	....	22	64	31,000	6200
12-45	125	148	58	60	197	.			. ..	.	22	63	31,000	6200
12-50	127	144	.	..	197	..	↓	↓	.	. . .	20	60	31,000	6200
12-52½	.....	..	.. .	.	195	123.2	318.2	....	Dam in	..	.....	.....	.....	.....
12-55	....	..	.....	. .	182	133.8	315.8	.....	4	.. ..	.....	58	30,000	6000
12-57½	.....	. . .	. ..	.	175	144.9	319.9	.. ..	. .	.	. ..	.....	.....	.....
1-00	.....	..	.	.. .	165	155.2	320.2		3		.....	58	27,000	5400
1-02½	.....	.. ..	. ..	. . .	155	164.2	319.2	.	Dam out	.	.. ..	.....	.... .	.....
1-05	.....	.	. .	. .	.	173.7		.	3	. .	....	58	25,000	5000
1-06	.....	. ...	. . .	..	146	175	321.0	.	.. ..	..	.....	.... .	..... .	.....
1-10	125	140	63	50	140	..	↑	↑	3	. .	.....	58	25,000	5000
1-15	125	140	63	45	140				4	..	28	57	24,000	4800
1-20	125	140	65	40	140	.			4	. .	29	59	25,000	5000
1-25	125	140	65	40	142	..			4	. ..	30	59	26,000	5200
1-30	133	140	62	40	144	. ..	Average=320		4	.....	33	59	27,000	5400
1-35	133	140	66	40	145	. ..		320−145=175 average depth	5	. ..	33	61	27,000	5 00
1-40	132	145	65	40	146	.. ..			5	.. .	34	63	28,000	5600
1-45	130	142	62	45	147	.....			5	... .	34	62	27,000	5400
1-50	132	140	67	45	147	.....			5	.....	33	60	27,000	5400
1-55	132	140	66	50	147	.....			4	.....	33	59	27,000	5400
2-00	132	140	66	50	147	.....			5	.....	36	59	26,000	5200
2-05	130	140	65	50	147	.....			5	.....	34	61	28,000	5600
2-10	132	145	62	50	147	....	↓	↓	4		.. ..	61	27,000	5400

TABLE 8. — (Continued)

Time	Steam pressure	Mean air pressure	Mean switch pressure	Return pressure	Depth: Index gauge (A)	Depth: Tape and float (B)	Depth: Sum A+B	Depth: Assumed by gauge	Gauge head over meter	Total lift	Main compressor: Rev. per cycle	Main compressor: R. p. m.	Gallons pumped: In 5 mintues	Gallons pumped: In 1 minute
2–15	.....	.....	.....	.....	140	181 1	321.1	.....	4 Dam in	.....	.....	60	27,000	5400
2–17½	.....	.....	.....	.....	130	189 2	322.2	.....	.....	.....	.....	.....	.....	.....
2–20	.....	.....	.....	.....	124	198 7	322 7	.....	4	.....	.....	62	26,000	5200
2–22½	.....	.....	.....	.....	115	206 5	321.5	.....	.....	.....	36	.....	.....	.....
2–25	.....	.....	.....	.....	106	215 0	321 0	.....	4	.....	.....	62	24,000	4800
2–27	.....	.....	.....	.....	100	222 0	322 0	.....	Dam out	.....	.....	.....	.....	.....
2–28½	.....	.....	.....	.....	96	225 0	321.0	.....	.....	.....	.....	.....	.....	.....
2–30	135	140	62	.....	93	.....	↑	↑	4	.....	.....	62	21,000	4200
2–35	135	140	61	30	86	.....			4	.....	43	62	21,000	4200
2–40	133	140	61	30	82	.....			4	.....	45	62	20,000	4000
2–45	132	145	65	30	84	.....			4	.....	42	64	20,000	4000
2–50	130	140	66	30	86	.....	Average=321 5	321.5−89 8=231 7 average depth	4	.....	44	64	21,000	4200
2–55	134	140	65	30	88	.....			4	.....	45	65	21,000	4200
3–00	135	140	62	30	90	.....			5	.....	44	67	22,000	4400
3–05	135	140	65	35	91	.....			5	.....	45	67	22,000	4400
3–10	135	140	62	30	92	.....			5	.....	43	66	22,000	4400
3–15	135	145	60	32	93	.....			4	.....	41	66	22,000	4400
3–20	135	140	61	28	94	.....			4	.....	41	65	22,000	4400
3–25	133	140	59	30	96	.....			4	.....	41	65	22,000	4400
3–30	132	140	61	30	97	.....	↓	↓	5	.....	.....	65	22,000	4400
3–31	.....	.....	.....	.....	.....	.....	.....	.....	Dam in	.....	.....	.....	.....	.....
3–35	.....	.....	.....	.....	90	231.5	321 5	.....	4	.....	.....	64	22,000	4400
3–37½	.....	.....	.....	.....	83	239 2	322.2	.....	.....	.....	.....	.....	.....	.....
3–40	.....	.....	.....	.....	77	246.0	323 0	.....	4	.....	.....	66	19,000	3800
3–42½	.....	.....	.....	.....	73	252.0	325 0	.....	.....	.....	.....	.....	.....	.....
3–45	.....	.....	.....	.....	67	255 5	322 5	.....	4	.....	.....	65	19,000	3800
3–47½	.....	.....	.....	.....	62	261 3	323 3	.....	.....	.....	.....	.....	.....	.....

TABLE 8. — (Continued)

3-50	...	...	...	...	57	265.8	322.8	...	4	...	...	66	17,000	3400
3-52½	...	...	...	...	53	270.0	323.0	...	...	...	...	...	...	...
3-55	...	...	...	...	50	273.8	323.8	...	4	...	...	69	16,000	3200
3-57½	...	...	...	...	43	278.0	321.0	...	...	...	...	...	...	...
4-00	...	...	...	...	40	282.0	322.0	...	4	...	...	67	15,000	3000
4-02½	...	...	...	...	36	286.5	322.5	...	...	...	...	...	...	...
4-05	...	...	...	...	34	288.5	322.5	...	4	...	...	68	14,000	2800
4-07½	...	...	...	...	30	290.8	320.8	...	...	...	...	...	...	...
4-10	...	...	...	...	28	294.0	322.0	...	4	...	...	68	13,000	2600
4-12½	...	...	...	...	26	295.8	321.8	...	...	...	...	...	...	...
4-15	...	...	...	...	25.5	296.8	322.3	...	3	...	...	58	10,000	2000
4-17½	...	...	...	...	24.5	297.4	321.9	...	Dam out	...	...	...	...	...
4-20	...	...	...	...	26	296.5	322.5	...	3	...	...	58	8,000	1600
4-22½	...	...	...	...	25.4	296.4	321.8	...	...	...	...	...	...	...
4-25	...	...	...	...	24.5	297.0	321.5	...	3	...	...	58	10,000	2000
4-27½	...	...	...	...	26	297.7	323.7	...	...	...	...	...	...	...
4-30	...	...	...	...	26	297.7	323.7	...	3	...	...	60	10,000	2000
4-32½	...	...	...	...	26	297.7	323.7	...	...	...	...	...	...	...
4-35	...	...	...	...	25	298.4	323.4	...	3	...	...	68	10,000	2000
4-37	...	...	...	...	24	299.4	323.4	...	Dam in	...	...	...	...	...
4-39½	...	...	...	...	23	301.0	324.0	...	Dam out	...	...	...	...	...
4-40	...	...	...	...	...	...			3	...	...	66	11,000	2200
4-45	135	135	58	10	24	...	Average=322.5	322.5−30.5=292 average depth	3	...	...	65	9,000	1800
4-50	133	133	58	10	26	...			3	...	70	66	11,000	2200
4-55	133	133	59	10	28	...			3	...	67	64	11,000	2200
5-00	132	132	58	10	30	...			3	...	71	64	11,000	2200
5-05	130	130	68	8	33	...			3	...	...	62	12,000	2400
5-10	132	132	68	8	33	...			3	...	75	62	11,000	2200
5-15	132	132	58	8	33	...			3	...	83	62	12,000	2400
5-20	135	135	58	8	33	...			2	...	...	64	12,000	2400
5-25	135	135	57	8	33	...			2	...	...	64	12,000	2400
5-30	135	135	60	8	32	...			2	...	92	66	10,000	2000
5-35	135	135	68	8	30.5	...			2	...	94	62	10,000	2000
5-40	...	...	...	...	31	...			2	...	92	63	10,000	2000
5-45	...	...	...	...	30.5	(297.5)			1	...	92	62	9,000	1800

TABLE 9
(March 23, 1904)

Time	Steam pressure	Mean air pressure	Mean switch pressure	Return pressure	Depth				Gauge head over meter	Total lift	Main compressor		Auxiliary compressor rev. per min.	Gallons pumped	
					Index gauge (A)	Tape and float (B)	Sum (A+B)	Assumed by gauge			Rev. per cycle	R. p. m.		In 5 minutes	In 1 minute
10-09¼	....	....	......											....	....
10-10	135	70	55		310									....	....
10-15	135	135	60	90	300							56		....	....
10-20	135	145	55	85	262	36 9	298 9	48	8		14	56		42,000	8400
10-25	135	150	54	85	228	71 0	299 0	82	7		15	57		39,000	7800
10-30	135	148	57	75	182	99 5	281 5	128	5		17	59		34,000	6800
10-32	135	......	......		170	111 0	281 0	140						....	....
10-34	135	....	....		162	121.3	283.3	148						....	....
10-35	135	145	60	70	....			......	5		21	62		32,000	6400
10-36	135	......	....		152	131.8	283 8	158							
10-38	135	......	.....		144	141 0	285 0	166						....	....
10-40	135	145	60	60	137	150 0	287 0	173	4 Dam	out				....	....
10-41¼	135	....	......		132	155.7	287.7	178						....	......
10-45	135	145	62	55	....			.....	4		27	65		27,000	5400
10-53	135	....	....		118	182 0	300.0	192							
10-54	135	.....	....		117	187 0	304 0	193						....	....
10-50	135	145	64	53	124	.....		↑	3 Dam in		27	63		28,000	5600
10-55	135	140	63	45	115	......			3 Dam out		29	62		26,000	5200
11-00	135	140	65	45	123	......	....		3		34	63	....	26,000	5200
11-05	135	140	65	45	125	....	...		3		34	62	....	25,000	5000
11-10	135	140	67	45	125	..	..	310−124=186 average depth	4		35	60	....	25,000	5000
11-15	135	140	68	45	124	...	...		5		35	66	64	25,000	5000
11-20	135	140	68	45	124	......	......		5 Blow	out	44	64	......	27,000	5400
11-25	135	135	67	45	128	......	......		3		.....	69	140	1,000	200
11-30	135	140	68	45	125	.....	.....		3		28	62	....	24,000	5800
11-35	135	140	68	45	125	...	.....		4		33	62	23	25,000	5000
11-40	135	140	68	45	124	.....	.....		3	....	36	67	51	25,000	5000
11-45	135	140	70	45	123	......	......		4	...	37	66	81	25,000	5000
11-50	135	140	68	45	124	.....	(310)	↓	3	..	39	67	79	26,000	5200

TABLE 9. — (Continued)

Time	Steam pressure	Mean air pressure	Mean switch pressure	Return pressure	Depth: Index gauge (A)	Depth: Tape and float (B)	Depth: Sum (A+B)	Depth: Assumed by gauge	Gauge head over meter	Total lift	Main compressor: Rev. per cycle	Main compressor: R. p. m.	Auxiliary compressor rev. per min.	Gallons pumped: In 5 minutes	Gallons pumped: In 1 minute
11-55	135	140	68	45	124			↑	4		40	65	75	27,000	5400
12-00	135	142	68	45	125				5		41	66	58	27,000	5400
12-05	135	142	67	45	125				5		42	65	33	26,000	5200
12-10	135	142	68	45	125			310−126=184 average depth	5		41	64	7	27,000	5400
12-15	135	142	68	45	125				5		39	68	11	26,000	5200
12-20	135	142	67	45	125				5		39	66	51	27,000	5400
12-25	135	142	67	45	126				5		39	66	30	27,000	5400
12-30	135	142	67	45	126				5		39	66	29	26,000	5200
12-35	135	142	68	45	126				5		38	62	38	25,000	5000
12-40	135	142	67	46	127				5		39	61	47	26,000	5200
12-45	135	140	68	47	128				5		38	61	77	26,000	5200
12-50	120	140	68	46	127		(310)	↓	5		38	60	52	24,000	4800
12-55	125	140	68	45	126			↑	5		40	62	28	27,000	5400
1-00	125	140	70	45	125				5		41	63	17	26,000	5200
1-05	130	140	70	45	124				5		40	65	14	26,000	5200
1-10	130	142	72	45	123 5				5		39	65	32	26,000	5200
1-15	133	142	70	45	123			310−125 5=184 5 average depth	5		41	67	31	26,000	5200
1-20	135	142	70	45	122				5		40	65	44	25,000	5000
1-25	137	142	70	43	121				5		40	65	56	26,000	5200
1-30	133	142	73	44	121				5		40	66	18	25,000	5000
1-35	134	142	70	45	121				5		40	67	58	26,000	5200
1-40	134	141	70	44	121				5		40	66	55	25,000	5000
1-45	134	141	71	44	122				5		40	67	32	26,000	5200
1-50	132	140	70	44	122		(310)	↓	5		41	69	37	26,000	5200
1-55	134	142	72	40	.. ..			. .	5		39	66	33	26,000	5200
1-58	....	....	....	.	110	213 4	323 4	. ..	. .	..	..	.. .	.. ....	....	
2-00	135	144	70	40	103	219 9	323 0	.. .	5	..	44	67	37	24,000	4800
2-02	....	....	. . .	. .	96	227 6	323 6	......	......	. .	... ..	......	....	....	....
2-04	....	.... ..	.. .. .	.. ....	90	233.5	323 5	......	... .	.. . .	. ....	......	... .	. ...	....
2 05	135	145	69	28	87	..	......	......	5	. ....	48	68	31	23,000	4600

TABLE 9. — (Continued)

Time	Steam pressure	Mean air pressure	Mean switch pressure	Return pressure	Depth				Gauge head over meter	Total lift	Main compressor		Auxiliary compressor rev. per min.	Gallons pumped	
					Index gauge (A)	Tape and float (B)	Sum (A+B)	Assumed by gauge			Rev per cycle	R. p. m.		In 5 minutes	In 1 minute
2-06					84	238 2	322 2								
2-08					79	244 2	323 2								
2-10	135	140	68	27	73	249 8	322 8		5		54	69	28	20,000	4000
2-12					68	254 7	322 7								
2-14					64	259 4	323 4								
2-15	135	143	73	23	62				5		57	70	26	18,000	3600
2-16					59 0	265 3	324 3								
2-18					55 5	268 8	324 3								
2-20	136	140	70	20	51	273 4	324 4		4		60	69	28	17,000	3400
2-22					46	277 8	323 8								
2-24					42	281 2	323 2								
2-25	135	147	72	14	41				4		65	70	25	15,000	3000
2-26					38	284 7	322 7								
2-28					35 5										
2-30	135	140	73		32				5		68	70	23	15,000	3000
2-32					28										
2-34					25	294 0	322 0								
2-35	135	140	73	10	24				5		73	73	17	12,000	2400
2-36					23	298 4	321 4								
2-38					21	298 9	319 9								
2-40	135	137	72	8	19 5		↑	↑	5		77	72	18	13,000	2600
2-45	137	137	71	8	20				5		77	75	18	11,000	2200
2-50	137	137	72	3	19			318−18 5	5		80	73	54	11,000	2200
2-55	135	137	70	3	18		318	=299 5	5		82	72	69	13,000	2600
3-00	137	135	70	3	18			average depth	5		85	72	72	12,000	2400
3-05	135	135	70	3	18				5		85	69	55	11,000	2200
3-10	136	137	71	3	18				5		82	71	68	12,000	2400
3-15	135	137	72	3	18		↓	↓	5		84		80	13,000	2600
3-18¼				Blow	out						95			11,000	220
3-55															

TABLE 9. — (Continued)

Time	Steam pressure	Mean air pressure	Mean switch pressure	Return pressure	Depth: Index gauge (A)	Depth: Tape and float (B)	Depth: Sum (A+B)	Depth: Assumed by gauge	Gauge head over meter	Total lift	Main compressor: Rev. per cycle	Main compressor: R. p. m.	Auxiliary compressor rev. per min.	Gallons pumped: In 5 minutes	Gallons pumped: In 1 minute
3-59					48										
4-00	137	137	70		47				5		52				
4-01					48	268 0	316 0							5,000	1000
4-03					46	271 0	317 0								
4-05	137	138	71	15	46	276 0	322 0		5		52			13,000	
4-07					42	278 7	320 7								2600
4-09					38	282 7	320 7								
4-10	135	138	71		37				5		59	66	136	13,000	
4-11					36	287 0	323 0								2600
4-13					32	289 2	321 2								
4-15	137	139	71	14	29	291 8	320 8		6						
4-16					Blow	out					76		91	14,000	2800
4-17½					28	294 6	322 6								
4-20	137				28 5										
4-21					28	292 3	320 3								900
4-23					29	292 0	321 0		5						
4-25	137	138	70	4	26	295 0	321 0		5		70			In 10 min	
4-27					24	298 0	322 0							9,000	900
4-30	137	137	73	4	21	300 8	321 8		4		70	68			
4-31					20	302 8	322 8							10,000	2000
4-33					18	304 0	322 0								
4-35	138	137	71	4	17	306 5	323 5				68	55			
4-37					16 5	306 6	323 1							6,000	
4-39					14										1200
4-40	135	138	71	3	13	309 0	322 0				74	61		←33,000 gal. in 28 minutes→	←1178 gal. in 1 minute→
4-42					12	313.0	325 0								
4-45	137	136	69	0	9						74	72	156		
4-50	137	136	72								74	71	43		
4-55	136	136	71								54	71	40		
5-00	133	137	70								59	71	34		
5-03½						(322 0)							140	Final Blo	w out

At this point slowed down long enough to allow men at meter to come up.

CHAPTER IV

THE AIR LIFT

In general, the system of pumping water from bored wells is termed the "air lift." This is by far the most common method of compressed air pumping. The air lift is found in municipal waterworks, ice factories, breweries, irrigation plants, dye works, cold storage and packing houses, and numerous other places the world over. In spite of its universal application there is little known by engineers concerning its proper design and installation. This is due to scarcity of literature on the subject; and then, too, the system is apparently so simple that at first sight it does not seem to merit the thought and analysis that almost every other proven mechanical appliance or device receives.

Historical. — We find in the book of Heron the first application of compressed air to lifting water. The arrangement is known as the "Fountain of Heron," both compressor and pump being combined in one system. Fig. 36 illustrates diagrammatically the ingenious principle of operation.

The air-tight vessels A and B are connected by pipes as shown. B is divided into two compartments C and D. Pipe a connects compartment C with the top of vessel A and pipe b connects the top of compartment D with the bottom of A. Pipe j is the discharge or eduction pipe. The operation is as follows:

Water is admitted into D through the pipe k and valve v until some predetermined level such as ef is reached when the supply is cut off by closing the valve. Water is next admitted to C through pipe w and flows down pipe a into A. As the water level in A rises, the air above the surface is compressed to a pressure corresponding to the height of the water column in a and C. This air pressure is exerted on the water surface ef forcing the contents of D out through the discharge pipe j.

The "Fountain of Heron" was employed about the middle of the 18th century in the mines of Chemnitz, Hungary. In 1797 some laboratory experiments were performed by a German mining engineer, named Loscher, on an air-pumping system of his own invention. His experiments are described in a pamphlet

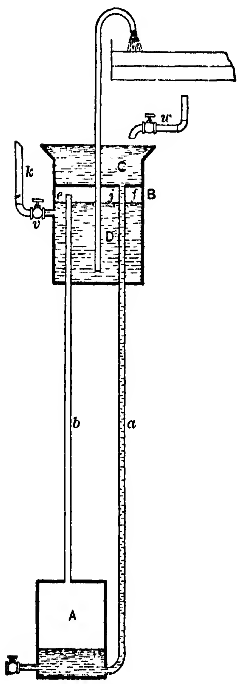


FIG. 36.

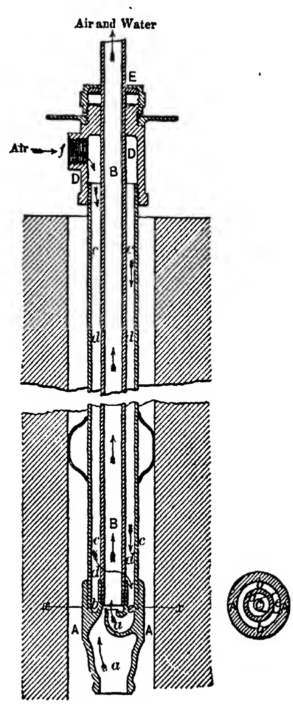


FIG. 37.

entitled *Aerostatiches Kunstgezeng.* Probably the first practical installation of the air-lift proper was made on some oil wells in Pennsylvania, about 1846, by an American engineer named Cockford. At about this same time Siemens, in England, experimented with the air lift, and in 1865 A. Brear patented what he terms an "oil ejector." In Fig. 37 is shown the arrangement

used by Brear. In letters patent No. 47793 he explains the operation of his invention. The principle is plainly indicated by the arrows in the illustration.

In 1880 J. P. Frizell was granted a patent (No. 233499) on a "New Method of Raising Water by Means of Compressed Air."

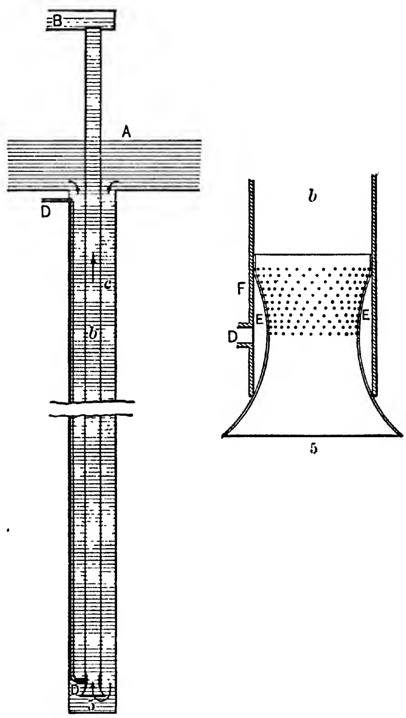


FIG. 38.

Fig. 38 shows this invention in detail, and it is described in the letters patent as follows: "My present invention has for its object the elevation of water in a simple and convenient manner by the introduction thereunder of compressed air; and it consists in causing a column of water to ascend in a pipe or conductor by the injection therein, at or near its bottom, of compressed air,

the weight of the air and water thus commingled being overcome by the weight of the external water, which is thus utilized as a motive power to elevate the water to the desired point. To enable others skilled in the air to understand and use my invention, I will proceed to describe the manner in which I have carried it out. In the said drawing, which represents, in section, my contrivance for elevating water, *A* denotes the surface of the body of water to be drawn from, and *B* the reservoir into which it is to be raised. *C* is a shaft or pit sunk in the earth to a depth corresponding to the pressure of the air used, and communicating with the body of water *A*. *b* is the rising pipe extending from near the bottom of the pit up to the reservoir *B*. The height from the surface *A* to the surface of *B* is the lift. Experiments show that the depth of the pit, reckoned from *A*, should be as much as five or six times the lift.

"Into the bottom of the rising pipe is fitted the hour-glass-shaped pipe 5, enclosing between the two pipes the annular space *EE*.

"The upper end of the pipe 5 is perforated with a great number of minute orifices, *F*, as indicated by the black dots. The lower end expands to a greater width than that of the rising pipe in order to diminish the resistance of the water in entering.

"The pipe *D*, leading from the source of compressed air, opens into the annular space *EE*. The pit or shaft *C* and rising pipe *b* being filled with water to the level of *A*, compressed air is admitted to the pipe *D* and passes into the annular space *EE*, thence through the perforations *F* into the water in the pipe *b*, through which it rises in the form of minute bubbles.

"The pipe *D*, which conveys the compressed air, may pass down in the pit *C*, as shown, or inside the rising pipe *b*, or outside the pit *C* in the ground if preferred.

"The pit or shaft *C* may, of course, be dispensed with if there is naturally a sufficient depth of water, it being merely necessary to introduce compressed air within the pipe or conductor, through which the water is to be raised at or near its bottom in order that it may rise, expand and diminish the weight of the column of water therein, as before described."

In 1884 another patent (No. 309214) was granted to S. S. Fertig on an annular tube pump. In 1885 Werner Siemens used an air-lift pumping arrangement in a mine shaft near Berlin, and in the same year Laurent used it for lifting sulphuric acid in France.

In 1892 Dr. Julius G. Pohlé was granted a patent (No. 487639) on an "air lift." Fig. 39 shows the Pohlé air lift which differs in action and principle from the Frizell only in the application of the compressed air. Dr. Pohlé describes his invention as follows:

"The object of the invention is to successfully and practically effect the elevation of the water to a much greater height than has heretofore been deemed economic with compressed air and to avoid the results due to an intimate commingling of the air and water, as well as to dispense with all valves, annular spaces and solid pistons. In accordance with my invention the air is not directed into the water in the form of fine jets or bubbles, which would very readily commingle intimately with the water, but is delivered in mass, and the water and air ascend in well-defined alternate layers through the eduction pipe.

"The drawings represent the apparatus in a state of action — pumping water — the shaded sections within the eduction pipe *W* representing water layers and the intervening blank spaces air layers.

"At and before the beginning of pumping, the level of the water is the same outside and inside of the discharge pipe *Q* incidentally; also in the air pipe. Hence the vertical pressures per square inch are equal at the submerged end of the discharge pipe. When, therefore, compressed air is admitted into the air pipe *a*, it must first expel the incidental standing water before air can enter the eduction pipe *W*. When this has been accomplished, the air pressure is maintained until the water within the eduction pipe has been forced out, which it will be in one unbroken column, free from air bubbles. When this has occurred, the pressure of the air is lowered, or its bulk diminished, and adjusted to a pressure just sufficient to overcome the external

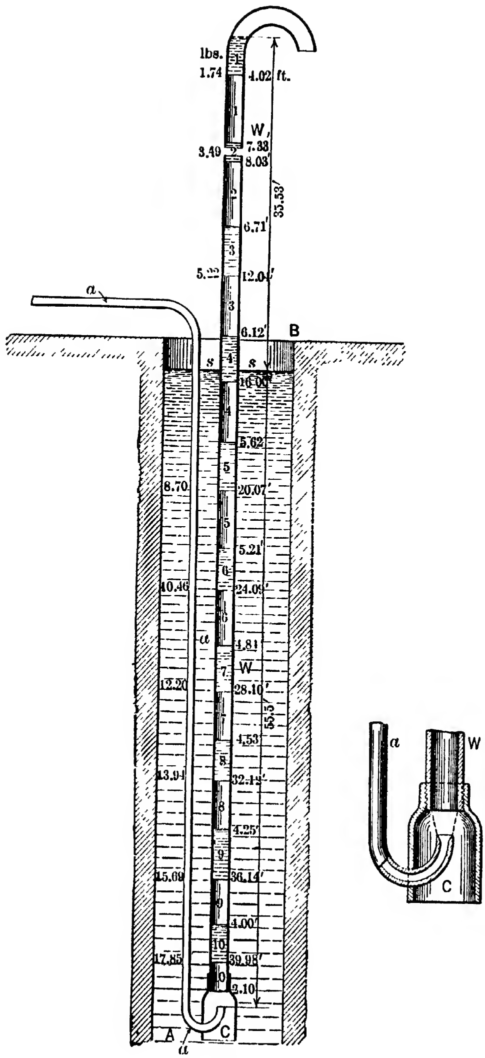

lbs.
1.74
4.02 ft.
W
7.33
3.49
8.03′
35.53′
6.71′
5.22
12.04′
a
6.12′
B
s
s
16.00′
5.62
8.70
20.07′
5.21′
10.46
24.09′
a
4.81
W
12.20
28.10′
55.5′
4.53
13.94
4.25′
15.69
36.14′
4.00′
17.85
39.98′
2.10
A
C
a
a
W
C

FIG. 39.

water-pressure. It is thus adjusted for the performance of regular and uniform work, which will ensue with the inflowing air and water which adjust themselves automatically in alternate layers or sections of definite lengths and weights. It will be seen in the drawings that the lengths of the water columns (shaded) and air (blank spaces) *I* and *I* are entered of the discharge pipe *W*, also that under the pressure of two layers of water, 1 and 2, the length of the air column 2 is 6.71 feet long, and so on. The lengths of aggregate water columns and the air columns which they respectively compress are also entered on the right of the water pipe. On the left of the water pipe are entered the pressures per square inch of these water columns or layers. Thus the pressure per square inch of column 1 is seen to be 1.74 pounds; that of 2, consisting of two columns or layers 1 and 2, each 4.02 feet long, to be 3.49 pounds, and that of 10, consisting of nine columns or layers of water 1 and 9, inclusive, each 4.02 feet long, and one 3.80, viz., layer 10 feet in length to be 17.35 pounds and the aggregate length of the layers of water is 39.98 feet in a total length of 91 feet of pipe. It will be noted that the length of pipe below the surface of the water in the well is 55.5 feet and that the difference between this and the aggregate length of the water layers (39.98) is 15.52 feet, that is, on equal areas the pressure outside of the pipe is greater than the pressure on the inside by the weight due this difference of level, which is 47.65 pounds for the end of the discharge pipe. It is the difference of 15.2 feet, acting as a head that supplies the water pipe, puts the contents of the pipe in motion and overcomes the resistance of the pipe. In general, the water layers are equal, each to each, and the pressure upon any layer of air is due to the number of water layers above it. Thus the pressure upon the bottom layer of air 10 in the drawings is due to all the layers of water in the pipe (17.35 pounds), and the pressure upon the uppermost layer of air 1 is due to the single layer of water 1 at the moment of its discharge beginning — viz., 1.74 pounds per square inch. As this discharge progresses this is lessened, until, at the completion of the discharge of the water layer, the air layer is of the same tension as the normal atmosphere."

In 1898 Mr. W. L. Saunders invented an air-lift pumping system in which air and water discharge takes place through a central pipe suspended from the well top. Referring to Fig. 40, compressed air is forced into the space between the discharge pipe

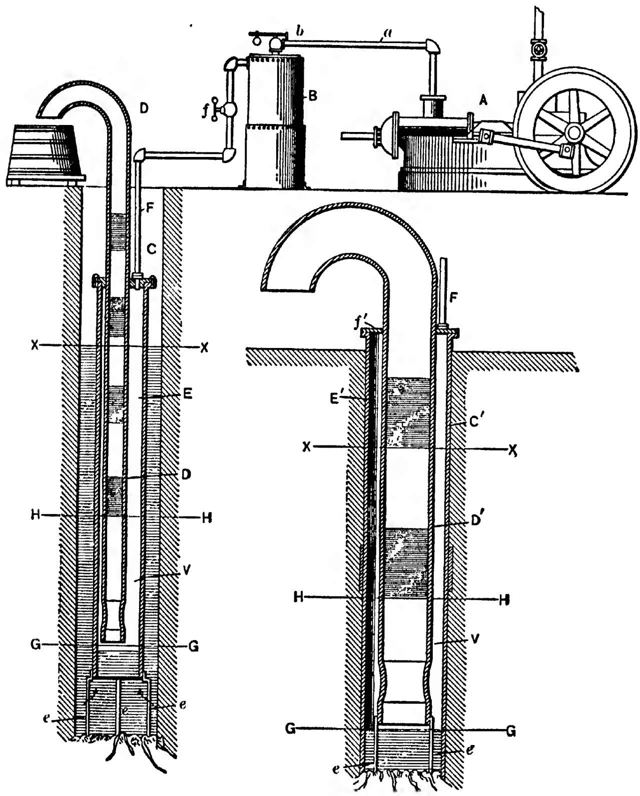


FIG. 40.

and well casing. This space is called the "pressure chamber." As the air pressure rises in the pressure chamber, the water level is forced downward until, finally, the end of the discharge pipe is uncovered when, immediately, compressed air enters the discharge pipe. This loss of air slightly lowers the pressure in the

chamber and the water rises in the chamber a distance equivalent to the pressure loss. The incoming air soon raises the pressure in the chamber, and the water head is again lowered beyond the end of the discharge pipe, when air again escapes, pressure reduction again occurs, and so on. Thus, it is claimed, alternate layers of air and water are formed which maintain their form until a point at or near the discharge is reached when a breaking up occurs.

Since 1898 a number of patents have been granted on special

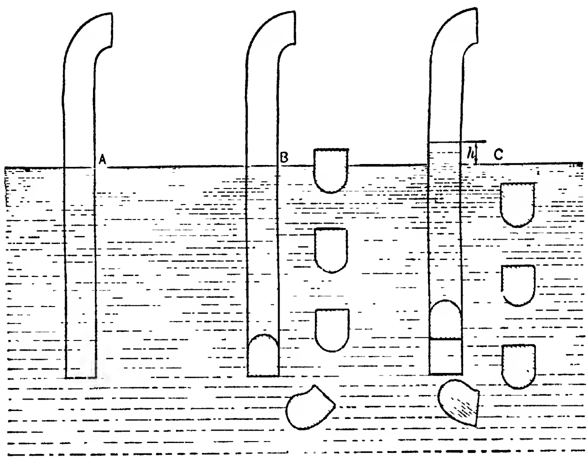


FIG. 41.

types and designs of foot pieces. Some of these are described and discussed on the following pages.

Principle. — The favorite method of illustrating the principle of the air lift is to assume a pipe open at both ends and partially submerged in a lake or other open body of water, as in *A*, Fig. 41. The water stands at the same level both inside and outside the pipe.

Assume now that a piston of air is forced down the outside of the pipe and up into the end, as in *B*, Fig. 41. This air piston displaces an equal volume of water in the pipe and, since air is

lighter than water, the hydrostatic pressure of the outside water column upon the lower end of the air piston is greater than the pressure due to weight of the air and water above on the inside of the pipe. This unbalanced condition causes the air piston to rise carrying the water before it until the level inside the pipe reaches a distance h (C, Fig. 41) above the level outside. The head h is equivalent to the difference between the weight of the air piston and the weight of the water it has displaced.

Another air piston admitted to the pipe end will, in the same way, cause the water in the pipe to rise higher. As more air pistons are admitted the water level will continue to rise until the upper pipe end is reached when overflow of air and water occurs. With a continued admittance of air pistons and continued overflow, there, obviously, always exists an unbalanced condition of pressures inside and outside the pipe which keeps up the discharge or overflow as long as air and water are provided at the lower pipe end.

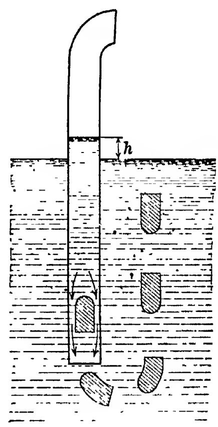


FIG. 42.

This will be recognized as the Pohlé description of the operation of an air lift. The air pistons mentioned by Dr. Pohlé do not, however, entirely fill the cross section of the pipe in actual practice. In other words, there is a space between the air piston and the walls of the pipe, which space is filled with water, as shown in Fig. 42. Each rising piston of air then does not carry all the water ahead of it, but some water escapes downward (with relation to the moving piston) around the piston. Thus, the air "slips" by a certain amount of water, and the loss which is known as slippage is the greatest to be contended with in this system of pumping.

In the Frizell system of operation small air bubbles are made to displace the water in the discharge pipe instead of larger air

pistons. A comparison of the Pohlé and Frizell systems is shown in Fig. 43. Both systems clearly depend upon a difference of pressures or, more correctly, specific gravities of the columns outside and inside the discharge pipe.

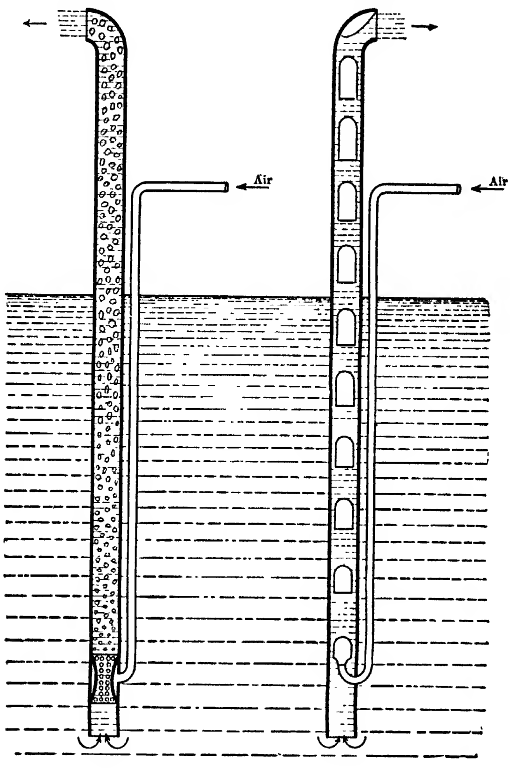


FIG. 43.

Air-lift Theory. — A number of attempts have been made to develop mathematical theories of the air-lift pump, but without satisfactory results. It is impossible to derive truly accurate formulæ expressing the air-lift theory because of the many uncontrollable variables met with. Probably the best theories

that have been advanced thus far are those of Prof. Elmo G. Harris in *Compressed Air*, and Dr. H. Lorenz in *Zeitschrift des Vereines Deutcher Ingenieure*, Vol. 53. Both of these discussions are given in full on the following pages.

Harris' Theory.* — "In Fig. 44, P is the water discharge or eduction pipe with area a, open at both ends and dipped into the water. A is the air pipe through which air is forced into the pipe P, under pressure necessary to overcome the head D. b is a bubble liberated in the water and having a volume O which increases as the bubble approaches the top of the pipe.

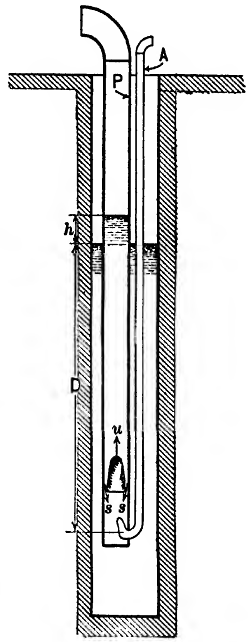


FIG. 44.

"The motive force operating the pump is the buoyancy of the bubble of air, but its buoyancy causes it to slip through the water with a relative velocity u.

"In one second of time a volume of water $= au$ will have passed from above the bubble to below it and, in so doing, must have taken some absolute velocity s in passing the contracted section around the bubble.

"Equating the work done by the buoyancy of the bubble in ascending, to the kinetic energy given the water descending, we have

$$wOu = wau\frac{s^2}{2g} \text{ where } w = \text{weight of water,}$$

or
$$\frac{O}{a} = \frac{s^2}{2g} \tag{35}$$

$\frac{s^2}{2g}$ is equivalent of the head h at the top of the pipe which is necessary to produce s, therefore $h = \frac{O}{a}$.

* Taken from *Compressed Air* by Prof. Elmo G. Harris.

"Suppose the volume of air O to be divided into an infinite number of small particles of air, then the volume of a particle divided by a would be zero, and therefore s would be zero; but the sum of the volumes O would reduce the specific gravity of the water, and, to have a balance of pressure between the columns inside and outside the pipe, the equation

$$wO = wah \text{ must hold.} \tag{36}$$

Hence again $h = \frac{O}{a}$, showing that the head h depends upon the volume of air in the pipe and not on the manner of its subdivision.

"The slip u of the air relative to the water constitutes the chief loss of energy in the air lift. To find this apply the law of physics, that forces are proportional to the velocities they can produce in a given mass in a given time. The force of buoyancy wO' of the bubble causes in one second a downward velocity s in a weight of water wau. Therefore,

$$\frac{wO}{wau} = \frac{s}{g} \tag{37}$$

whence $$u = \frac{O}{a}\frac{g}{s}$$

But $$\frac{O}{a} = \frac{s^2}{2g}$$

as proved above.

Therefore $$u = \frac{s}{2} = \sqrt{\frac{O}{a}\frac{g}{2}} \tag{38}$$

"This shows that the slip varies with the square root of the volume of the bubble. It is, therefore, desirable to reduce the size of the bubbles by any means possible.

"If $u = \frac{s}{2}$, then the bubble will occupy half the cross section of the pipe. This conclusion is modified by the effect of surface tension which tends to contract the bubble into a sphere. The law and effect of this surface tension cannot be formulated nor can the volume of the bubbles be entirely controlled. Unfortu-

nately, since the larger bubbles slip through the water faster than the small ones, they tend to coalesce; and, while the conclusions reached above may approximately exist about the lower end of an air lift, in the upper portion where the air has about regained its free volume, no such decorous proceeding exists; but, instead, there is a succession of more or less violent rushes of air and foamy water."

Lorenz's Theory.* — "Let

p_1 = the pressure in the foot piece;
p_b = the barometric pressure acting on the surface of the water in the well, and also on the discharge end of the pipe A, Fig. 45;
u_w = the density of the fluid pumped;
w_w = the weight per second of the water pumped;
w_a = the weight per second of the air discharged through the pipe B;
u_i = the density of the air at the air inlet in the foot piece;
u_b = the density of the air at the discharge end of pipe A;
v_1 = the velocity of the liquid in the pipe A below the air inlet;
c_e = the coefficient of entrance;
h_s = the depth of submergence.

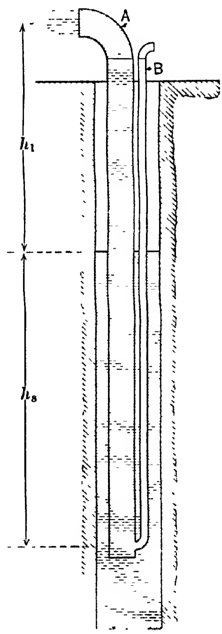


FIG. 45.

"Referring to Fig. 45, it may be seen that, during the operation of the pump, the following equation of heads holds between the pump and a point at the same elevation outside the pump:

$$h_s - \frac{p_1 - p_b}{u_w} = \frac{v_1^2}{2g}(1 + c_e) \tag{39}$$

* Taken from *An Investigation of the Air Lift Pump* by Profs. Davis and Weidner.

"For flow in the discharge pipe A, the following differential equation holds on account of the variable value of the density u of the mixture of gas and liquid:

$$dh - \frac{dp}{u} = \frac{v\,dv}{g} - cv^2\,dh \tag{40}$$

in which v equals the variable velocity, and p, the pressure at any point; and with the variable specific weight of air as u_v equation,

$$\frac{w_a + w_w}{u} = \frac{w_a}{u_v} + \frac{w_w}{u_w} \tag{41}$$

designates the momentary volume of the mixture $w_a + w_w$.

"If the mixture of air and fluid is very intimately commingled; that is, if the air penetrates the fluid in the form of small bubbles, it can be assumed that the air expands isothermally, so that

$$u_v = \frac{pu_b}{p_b} \tag{42}$$

By means of equations (41) and (42) the fundamental formula (40) becomes:

$$dh - \frac{w_a}{w_a + w_w}\frac{p_b\,dp}{u_b p} - \frac{w_w}{w_a + w_w}\frac{dp}{u_w} = \frac{v\,dv}{g} - cv^2\,dh \tag{43}$$

"Integrating this equation between the limits $h_s + h_l$ and o, p_i and p_b and v_i and v_b (the velocity of the mixture at the discharge end of the eduction pipe) there results:

$$- (h_l + h_s) + \frac{w_a}{w_a + w_w}\frac{p_b}{u_b}\log_e\frac{p_i}{p_b} + \frac{w_w}{w_a + w_w}\frac{(p_i - b_b)}{w_w}$$
$$= \frac{v_b^2 - v_i^2}{2g} + \int_o^{h_l+h_s} cv^2\,dh \tag{44}$$

"Replacing the last term in this equation, for the sake of simplicity, by assuming a mean coefficient c_p so that

$$\int_0^{h_l+h_s} cv^2\,dh = c_p\frac{l}{d}\frac{v_b^2}{2g} \tag{45}$$

there results

$$- (h_l + h_s) + \frac{1}{w_a + w_w}\left(w_a\frac{p_b}{u_b}\log_e\frac{p_i}{p_b} + \frac{w_w}{u_w}(p_i - p_b)\right) =$$
$$\frac{v_b^2 - v_i^2}{2g} + c_p\frac{l}{d}\frac{v_b^2}{2g} \tag{46}$$

Adding this equation to (38) gives

$$\frac{w_a}{w_a + w_w}\left(\frac{p_b}{u_b}\log_e\frac{p_i}{p_b} - \frac{p_i - p_b}{u_w}\right) = h_l + \frac{v_b^2}{2g}\left(1 + c_p\frac{l}{d}\right) + \frac{v_i^2}{2g}c_e \quad (47)$$

"Neglecting the second term on the left-hand side of the equation, which will be very small in comparison with the first term on account of the large difference between the values of u_w and u_b, w_a and w_w, and neglecting w_a in the denominator, this equation reduces to the simple form

$$\frac{w_a p_b}{w_w u_b}\log_e\frac{p_i}{p_b} = h_l + \frac{v_b^2}{2g}\left(1 + c_p\frac{l}{d}\right) + \frac{v_i^2}{2g}c_e \quad (48)$$

"In developing this energy equation, Dr. Lorenz assumed the velocity of air entering the foot piece as equal to that of the water; that is, free from any losses due to impact, which may be readily assumed on account of the small kinetic energy of the air. Now let

$$l_i = w_a\frac{p_b}{u_b}\log_e\frac{p_i}{p_b} \quad (49)$$

the work of isothermal expansion of the weight of air w_a, and $l_o = w_w h_l$ the work done in lifting the fluid weight w_w, from which the hydraulic efficiency

$$e = \frac{l_o}{l_i} = \frac{w_w h_l u_b}{w_a p_b \log_e\frac{p_i}{p_b}} \quad (50)$$

can be computed with the aid of equation (47)

$$\frac{1}{e} = 1 + \frac{v_b^2\left(1 + c_p\frac{l}{d}\right) + v_i^2 c_e}{2gh_l} \quad (51)$$

For the practical use of these formulæ it will be better to eliminate the velocities v_b and v_i by introducing the volumes q_b and q_w of weights w_a and w_w and using the area of the discharge pipe a_p, by means of the following formulæ:

$$\begin{aligned} w_a &= q_b u_b & \qquad w_w &= q_w u_w \\ q_b + q_w &= a_p v_b & \qquad q_w &= a_p v_i \end{aligned} \quad (52)$$

Writing now in place of equation (47)

$$\frac{q_b p_b}{q_w u_w} \log_e \frac{p_i}{p_b} = h_l + \left(\frac{\left[1 + c_p \frac{l}{d} \right] [q_b + q_w]^2 + c_e + q_w^2}{2\, g a_p^2} \right) \qquad (53)$$

and differentiating this equation with respect to q_b, putting

$$\frac{dq_w}{dq_b} = 0$$

there results as a requirement for maximum discharge q_w

$$\frac{1}{q_w u_w} \frac{p_b}{} \log_e \frac{p_i}{p_b} = \frac{1 + c_p \frac{l}{d}}{g a_p^2} (q_b + q_w) \qquad (54)$$

or in connection with equation (39), that is after eliminating the pressure p_i and p_b,

$$\left(1 + c_p \frac{l}{d} \right) (q_b^2 - q_w^2) = 2\, g h_l a_p^2 + c_e q_w^2 \qquad (55)$$

"If, now, the maximum discharge determined from the capacity of the well, and the area a_p of the discharge pipe determined from the diameter of the well, and also the lift and the known coefficients c_e and c_p are given, the volume of free air required may be computed by means of the formulæ (54) and (55), from which the submergence h_s can then be computed by means of equation (39). For these fixed conditions equation (53) then gives the relations between any desired values of q_b and q_w using the same pressure p_i."

There have been several other theories published but the formulæ presented are not so simple or so easily applied as those just given. In neither of these theories, however, has account been taken of the air-friction loss in the air pipe, or the losses due to curvature in the elbow at the well top, or entrance losses in the lower end of the discharge pipe. The formulæ may be easily modified to include these losses by applying principles given on following pages under curvature, entrance and air-friction loss headings.

CHAPTER V

SUBMERGENCE

In presenting this essential to the air lift, it is first necessary that the bored well itself be discussed briefly. A bored well is merely a cylindrical-cased hole of sufficient depth to penetrate the water-bearing stratum, and provided at the lower end with suitably located openings for free entrance of water from the stratum into the well.

Water is admitted to the well in two ways. One way is to attach a screen or strainer to that part of the well casing which is in the water-bearing stratum. This strainer consists of a piece of wire-wrapped perforated pipe (as shown in Fig. 46) equal in length to the depth of the stratum. Water finds its way through the openings between the wire strands, but sand, gravel and other foreign matter is held in the stratum. The other method (Fig. 47) is used when there is no sand present and where adequate anchorage, such as cap rock, immediately overlies the water-bearing stratum. As shown, the casing is resting on the rock, and the bare hole is continued through the rock into the stratum below. The water, then, is free to rise up in the well from the bottom.

The water-bearing stratum itself is a porous bed of sand, gravel, or it may even be rock formation, through which water flows. The ideal arrangement is the porous stratum lying between two impervious strata and thus confining the water flow to its bed with no escape either upward or downward. No such perfect formation is ever found, however, for there always exists fissures or openings in either or both confining strata through which water will escape. Nearly impervious confining strata are sometimes found, so in our discussion we will assume for simplicity, that all water is held within a pervious bed.

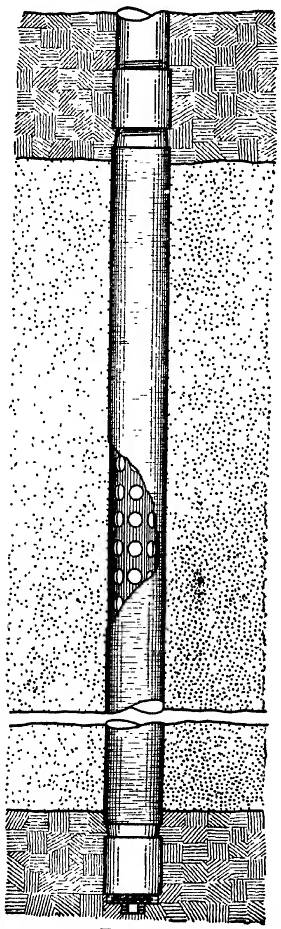
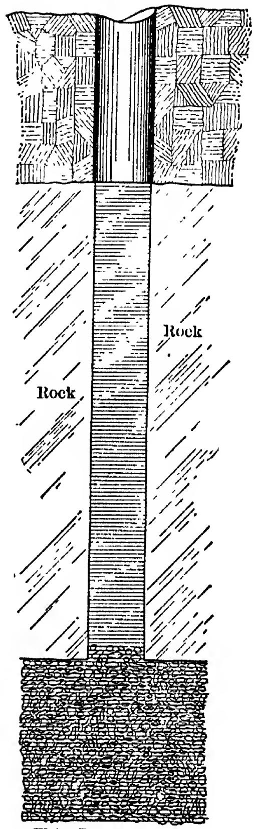


Fig. 46.



Water Bearing Stratum

Fig. 47.

At some distant point, the strata reach the earth's surface where the pervious one receives its water supply from rainfall, springs, rivers, etc. The principle is shown graphically in Fig. 48. A bored well, piercing the upper strata and entering the water-bearing stratum, is shown at A. The strainer is also shown. Water enters the stratum from the source of supply, as indicated by the arrows, and rises up inside the well to a level which is the same as that of the headwater, or BC in Fig. 49. These are the static conditions, and the distance below the ground level that the water stands is known as the *static head*.

The static head of a bored well, then, depends upon the water level in the source of supply and, when the latter is higher than the mouth of the well, a natural flow is obtained.

Consider, now, a bored well (Fig. 49), piped for operation with air and having a static head h, and a submergence h_s. The air pressure necessary to start the flow from the well is equivalent to the depth of submergence, or $.434\ h_s$. This is evident because the resistance that must be overcome is that offered by the vertical column of water standing over the air nozzle A. As air under pressure $.434\ h_s$ (plus friction) is furnished by the compressor, the water column is raised in proportion until, finally,

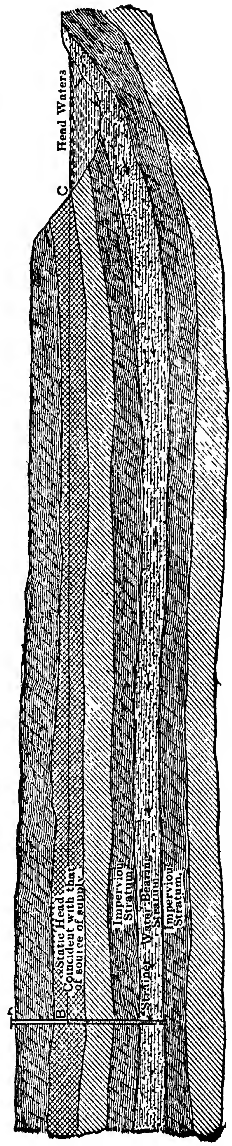


Fig. 48

the water surface just reaches the point of discharge. Suppose a valve were closed in the air line. There would be no further movement of the water column, and the discharge pipe from the air nozzle to the lower end of the water column is filled with compressed air while the remainder is filled with water. Suppose now that the air valve were opened and more compressed air admitted. Immediately water overflows the discharge pipe and the column is shortened. This decreases the resistance and, consequently, the compressed air behind the water column expands and the pressure drops. Thus the first head is blown from the well.

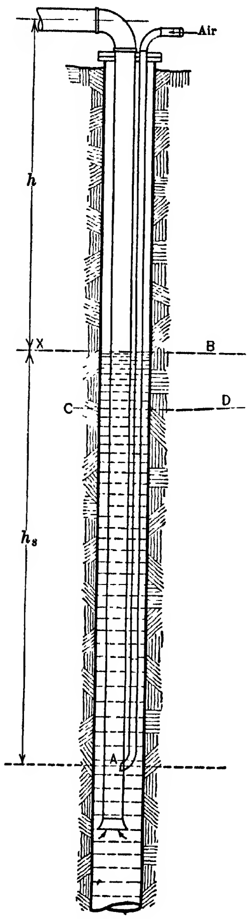


FIG. 49.

When the air pressure has become sufficiently reduced by the removal of the first column as explained, the hydrostatic pressure of the water head outside the well overbalances the internal resistance and water is forced into the well. This water in passing the air nozzle becomes aerated and continues to rise, endeavoring to balance the outside water-column pressure. Such a balance is impossible because of the reduced weight of the inner column, consequently a continuous overflow occurs at the discharge pipe end and operation has begun and continues as long as sufficient air is furnished.

The air pressure necessary to keep a well in operation is sometimes considerably lower than that necessary to start the well. This is due to two causes: first, a pressure drop due to established column momentum, and second,

a pressure drop due to actual falling of water level in the well.

To explain the first cause it suffices to state that less energy or pressure is necessary to keep a column of water moving than is required to start the same column when at rest. The pressure difference is equal to the velocity head of the moving column, or $.434 \frac{v^2}{2g}$.

The second pressure drop is considerably greater than the first mentioned and varies almost for every well. (Refer to Fig. 48 and note the static conditions as there shown.) When the well *A* is pumped, there is created a flow of water from the source *C* to the discharge pipe end. Immediately a loss of head appears in the stratum due to friction just as a friction loss occurs in a pipe line when transmitting water. The water column at the source, then, under dynamic conditions, cannot maintain an equal head in the well and, consequently, the dynamic head in the well is lower than the static head by an amount equal to friction loss between source and discharge. This means that the column of water over the air nozzle in the well is shorter by the same amount, hence the air pressure is correspondingly reduced.

The friction losses in water-bearing strata are governed by the same laws as those governing pipe-friction losses. The well drop, then, depends upon the resistance offered to the travel of the water by the obstructions in the stratum; upon the resistance offered by the strainer (or entrance loss at the well end if no strainer is used); upon the friction loss in the water discharge pipe and upon the amount of water being pumped. Clearly the head drop is different for each well and for each quantity of water pumped from any one well, and can neither be estimated nor otherwise determined except by experiment. By noting the starting and running air pressures and correcting for $\frac{v^2}{2g}$, the drop in head in feet may be determined by multiplying the pressure difference by 2.31.

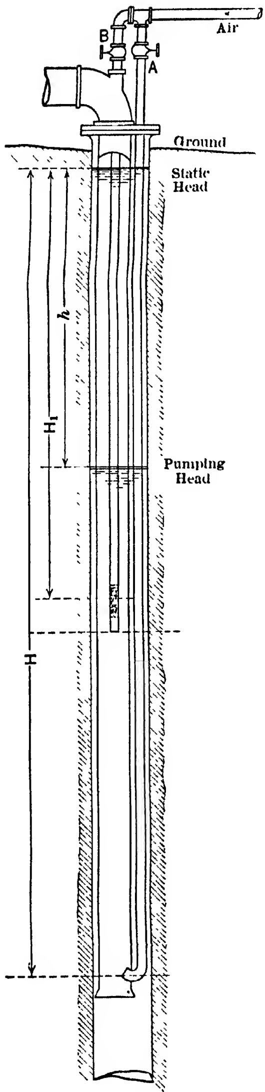


FIG. 50.

In some instances the head drop is slight, but in others it is excessive, and so much so that the starting pressure necessary is considerably higher than the compressor can possibly afford. To bring the starting pressure within the limit of the compressor, an auxiliary air line is often necessary, as shown in Fig. 50. The first head is pumped off by closing valve *A* and opening valve *B*, thus forcing the air through the shorter air line. After starting the well, the valve *A* is opened and valve *B* is closed. The starting pressure necessary is reduced from .434 H to .434 H_1, the former being necessary if only one air line were used.

In designing an air lift to meet any given set of conditions, it is necessary that the head drop of the well be first known, otherwise a proper proportioning of submergence is impossible. It is the uncertainty of the head drop more than any other one thing that makes each well an individual problem to be solved.

Having determined the head drop in any specific case when the desired quantity of water is being pumped, the next question is, what is the most economical submergence? Submergence governs pressure. Increased submergence means higher operating air pressure but decreased

air-slippage losses and consequently smaller air volume; decreased submergence means lower operating air pressure but greater air-slippage losses and, therefore, larger air volume necessary. Evidently there is a point where the energy expended or pressure through volume is least. This point may be found experimentally by varying the submergence and running a test on each change. When this is done, and the air nozzle set at the proper point, the over-all efficiency (which includes upkeep and repairs) of an air-lift pumping plant compares most favorably with that of any other means of deep-well pumping.

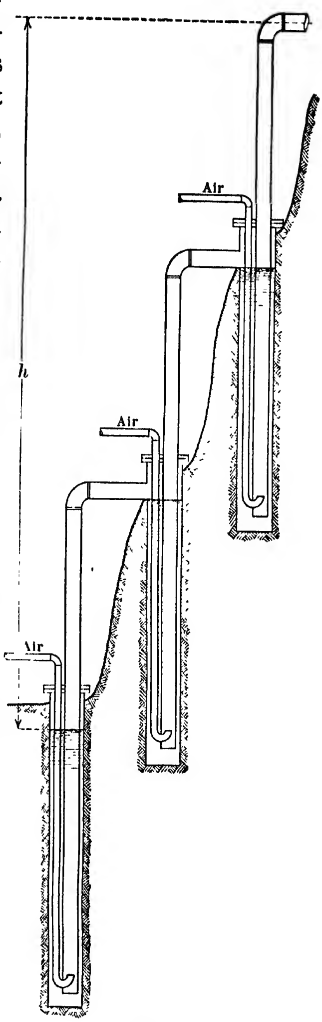


FIG. 51.

The high percentage of submergence necessary is one of the most serious drawbacks to the air lift. For operation in places other than in a well, lack of submergence is sometimes compensated for by dividing the lift into a number of stages in a similar manner as is done in high-duty pumping with the displacement pump, previously described in Chapter II. Fig. 51 shows diagrammatically the general idea of a stage lift. Owing to the small area available such an arrangement is impracticable inside of a bored well.

When two or more wells are to be operated together, their location with respect to one another will have an important bearing upon the ultimate results obtained. Referring to Fig. 50 the line *XP* is the surface of the water plane when the well is not being operated and *CD* is the surface when operating the well. The line *CD* connects the dynamic head in the well with the head at the source of water supply, and consequently inter-

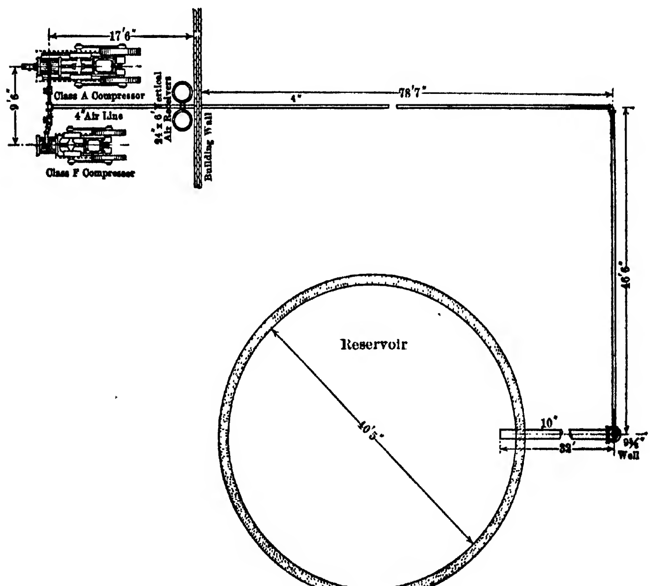
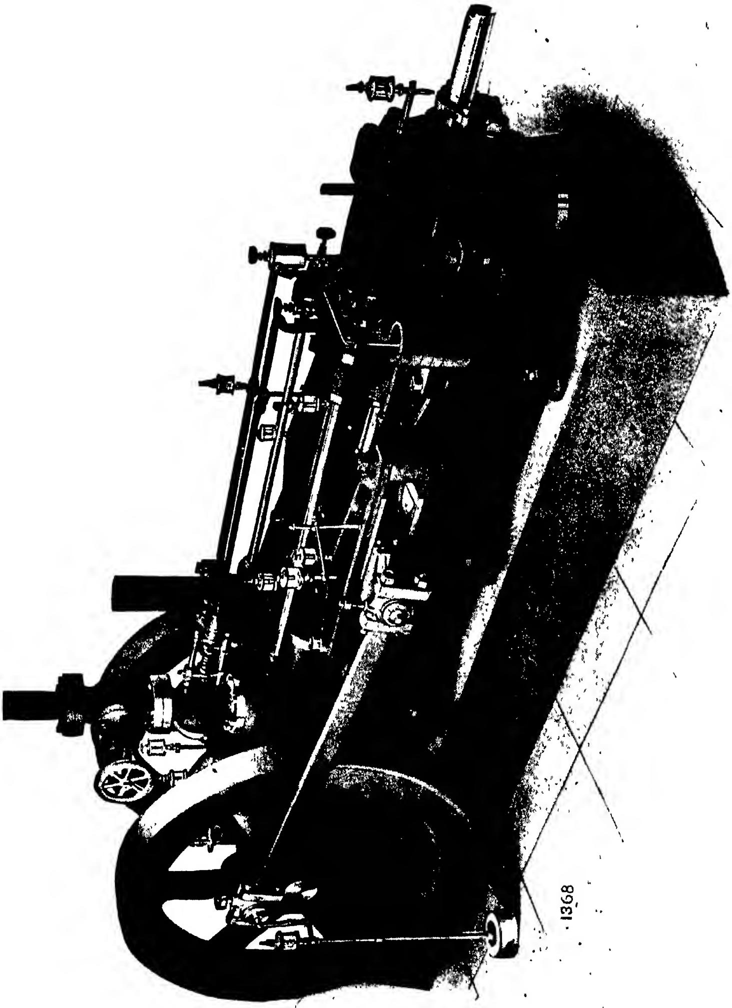


FIG. 52.

sects *XB* at the latter point. To locate the wells in a line parallel to the water flow would mean that each well would be affected by the drop in head of the other, and, consequently, the combined duty when pumping all wells would be unnecessarily high.

Again, the nearer together the wells are drilled, the greater will be their effect upon one another when operating. The


1368

most advantageous disposition of wells, then, is in a line at right angles to the direction of the subterranean flow and as far apart as possible.

Test. — To show the effect of varying submergence upon the pumping efficiency of an air lift, as well as to illustrate the steps necessary to properly proportion the piping, a test of a well owned by the city of Hattiesburg, Miss.,* is given on the following pages.

General Remarks. — Here, as is usually found, the city required all the water that could be obtained for the least cost, or, in other words, the most efficient capacity of the well. To determine this capacity, it was necessary to vary the submergence and run a test on each change, making the various observations shown on the log of results following. The air volume and pressures, of course, varied, and the compressor speed was so regulated in each trial that air neither was wasted by blowing through the water, nor was an insufficient quantity furnished which would cause the well to flow in "heads." A little experimenting before each trial showed just when the minimum air volume for steady operation was being supplied.

Equipment. — The equipment consisted of one 12″ and 12″ by 12″ Ingersoll-Sergeant Class "F" and one 14″ and $14\frac{1}{2}$″ by 14″ Class "A" compressors, two 24″ by 6 foot vertical air receivers fitted with gauges, safety valves, etc., and one $9\frac{5}{8}$-inch well with all necessary piping. Fig. 52 is a layout of the plant showing the location of the well, reservoir, etc., as well as the surface line lengths, and sizes. Figs. 53 and 54 are cuts of the compressors used.

The Well. — Fig. 55 shows the well. It will be noted that two water-bearing strata were tapped and a strainer located in each. It afterwards developed that the lower stratum was of little or no value.

The Foot Piece. — At the end of the air line in the well, the foot piece shown in Fig. 56 was attached. *aa* are wire nails which held the flanges apart as well as maintained the foot piece in a central position in the well. The object of the inverted oak

* Part of testimony in law suit Layne & Bowler Co. vs. City of Hattiesburg, Miss.

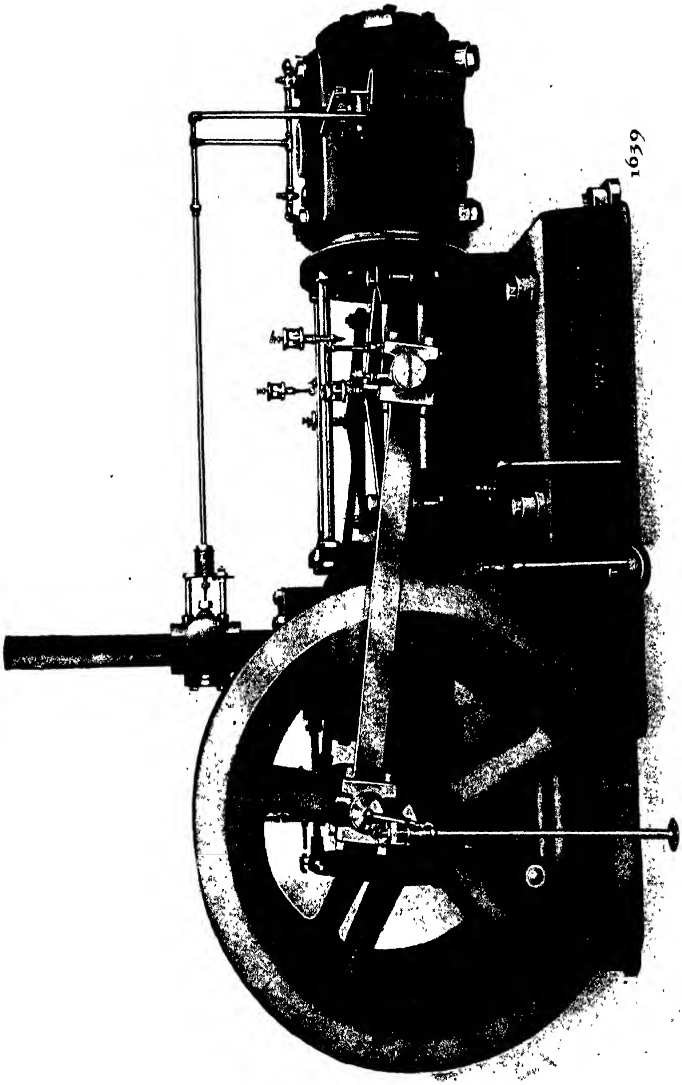


FIG. 54.

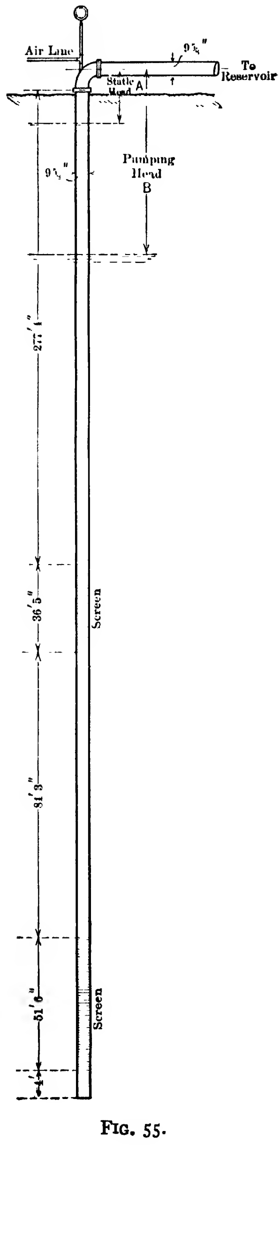
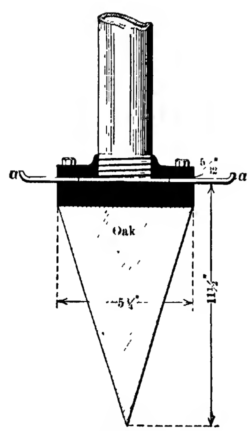


FIG. 55.



CROSS SECTION
OF
FOOT PIECE

FIG. 56.

cone attached to the lower half of the flange was to eliminate the eddy losses of the rising water column when striking the flanges.*

General Data. — The following dimensions, etc., were constant throughout the entire test: -

Total depth of well, feet	$453\frac{1}{2}$
Inside diameter of casing, inches	$9\frac{5}{8}$
Area of casing, square inch	72.95
Inside diameter of air line in well, inches	2.468
Outside diameter of air line in well, inches	2.875
Outside area of air line in well, square inch	6.492
Inside diameter of eduction pipe, inches	$9\frac{5}{8}$
Net area of eduction pipe (square inch)	66.267
Mean inside diameter of reservoir, feet	$40\frac{5}{12}$
Area in square feet	1282.98
Cubic feet of water per inch of depth	106.915
U. S. gallons per cubic foot	7.481
U. S. gallons per inch of depth	799.83
Net piston displacement per revolution of Class "A" compressor, cubic foot	2.486
Size Class "A" inlet pipe, inches	$3\frac{15}{16}$
Size Class "A" piston rod, inches.	$2\frac{15}{16}$
Size Class "F" piston rod, inches.	$1\frac{11}{16}$

Method of Procedure. — The air line length was varied, air volume regulated and a test of four, and sometimes six, trials made for each change. The efficiencies, etc., were computed as shown in Table 10. Each trial marked on the table shows results averaged from the several actual trials made.

During each trial, readings were taken of the boiler pressure gauge, air gauges at the receiver and well top, and of the thermometer registering the temperature of the atmosphere in the engine room near the compressor intake. Indicator diagrams were taken from the air cylinders and the amount of water being pumped was measured in the storage reservoir located just outside of the building.

In Fig. 57 are given superimposed indicator diagrams. These show the energy expended during each trial and the pressure of the air furnished. It is interesting to note the varying volumetric efficiencies and the difference in power required to lift

* Foot-piece designed by Layne and Bowler Co.

TABLE 10.

Mean Results of Tests of Water Well Owned by City of Hattiesburg, Miss.

Trial number	Rev. per min. of compressor	Cubic feet of piston displacement per minute	Volumetric efficiency	Temperature of intake air, deg. Fahr.	Cubic feet of free air per minute	Gallons of water per minute pumped	Cubic feet of water per minute	Boiler pressures	Air pressures (gauge): Starting pressure of well	Air pressures (gauge): (Operating) from the indicator diagrams	Air pressures (gauge): (Operating) at receiver	Air pressures (gauge): (Operating) at well top	Static head in well	Total pumping head	W.H.P.	A H P.	Length of air line, feet	Submergence, feet	Submergence, per cent	Pumping efficiency
1	150	372.9	94.5	88°	331.9	1495.7	199.8	100	96.25	80.8	78	77.5	3′ 7″	47.5	17.9	55.0	224	177.5	79	32.6
2	155	385.3	95.6	79°	355.3	1459.1	193.5	105	87.35	73	70	69.5	3′ 11″	45.8	16.9	53.6	204	159.1	76.5	31.5
3	156	389.1	96.5	81°	359.1	1419.7	189.8	100	78.5	65	62	61.5	4′ 0″	44.3	15.9	51.1	184	140.4	76	31.1
4	152	377.9	96.5	85°	345.8	1359.7	181.6	100	69.2	56.5	54.5	54.5	3′ 11″	39.4	13.5	45.5	162	123.6	75.5	30.0
5	151	376.0	96.7	87.5°	342.9	1299.7	173.7	107	60.8	50.0	46.5	46.0	3′ 1″	37.7	12.4	40.6	142	105.9	74.5	30.6
6	147	365.4	96.7	88°	332.9	1219.7	163.2	105	52.6	41	38.5	38.0	3′ 7″	37.5	11.55	35.2	124	86.6	69.5	32.8
7	140	349.2	96.7	83°	321.7	1105.8	147.5	100	44.25	33	30.5	30.0	3′ 10″	37.1	10.4	28.3	105	69.2	65.5	36.7
8	135	335.6	97.5	77.5°	315.3	1007.8	134.7	100	36.5	29	24.5	24	3′ 6″	32.5	8.26	24.75	86.5	54.3	62.5	33.4
9	126	313.8	97.5	85°	290.8	903.8	120.5	100	32.8	26	22	22	4′ 0″	29.3	6.65	21.4	79	49.9	63	31.25
10	115	285.2	98.2	86°	267.6	802.3	107.1	105	27.5	21	18	13	4′ 0″	26.0	5.29	17.5	67	40.7	60.9	30.5
11	105	260.4	98.2	86°	242.8	690.5	92.3	100	16.5	13.5	10.5	10.5	4′ 0″	21.2	3.71	12.8	42.5	21.2	50	29.8

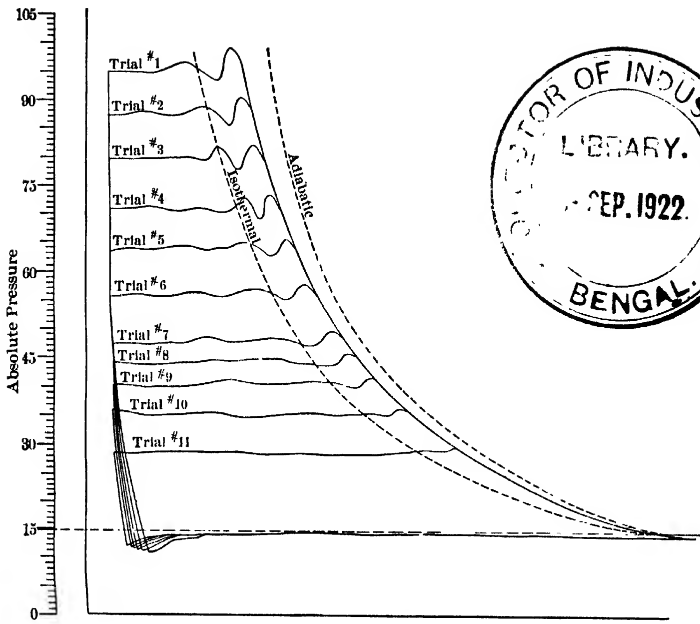


FIG. 57.

the discharge valves when operating against various terminal air pressures.

In Fig. 58 is a curve plotted between well head drop in feet as the ordinate and gallons of water per minute as the abscissa. The curve shows a fairly regular drop between 700 G.P.M. and 1100 G.P.M., but as the quantity is increased further, the curve is broken and irregular. This is due to the fact that it was necessary to force the well beyond its normal capacity to yield greater than 1100 G.P.M.

Referring to Table 10, it is seen that 1106 gallons of water per minute is the economical capacity of the well, and that the pumping head is 37.1 feet. The next thing to be determined is the submergence best suited. In other words, when pumping the economical capacity of the well, and hence operating against the

corresponding pumping head, how far below the 37.1 feet should the air nozzle be located? This can readily be determined by again changing the length of the air line, and so regulating the compressor speed at each change that the quantity pumped from

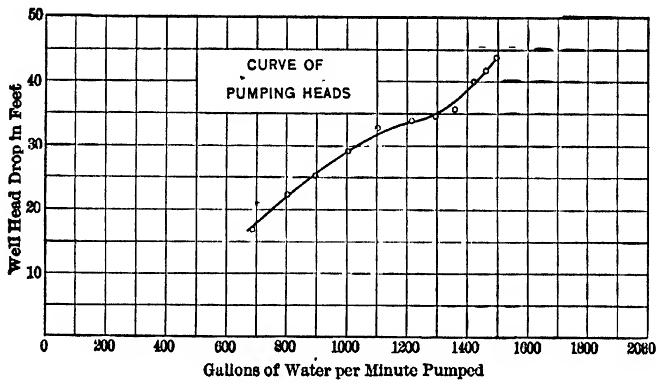


FIG. 58.

the well will remain the same, *i.e.*, 1106 gallons of water per minute. Tests run as before will now show the most economical point of submergence when pumping this desired quantity of water. Usually five or six such tests will suffice and the results plotted on coördinate paper will give the proper location of air line. Nine trials were made on the Hattiesburg well as shown in Table 11, which is the log of results and the computations. Fig. 60 is the submergence curve plotted from these results and is typical of such curves.

Figure 59 is a photograph of the discharge from the well after making the final submergence adjustments for the economical capacity of the well. As will be observed, there are small air slippage losses and the flow is remarkably constant for an air-lift.

The submergence curve shows that, between 65 per cent and 50 per cent and between 75 per cent and 95 per cent submergence,

the efficiency falls off very rapidly, but between 65 per cent and 75 per cent submergence, the efficiency difference amounts to less than 1 per cent. By extending the curve downward on the left, it is shown to be impracticable to pump at all under 20 per cent submergence. It was impossible to test under lower submergence than 50 per cent because of insufficient available air

Fig. 59.

TABLE 11

MEAN RESULTS OF SUBMERGENCE TESTS OF WATER WELL OWNED BY CITY OF HATTIESBURG, MISS.

Trial number	Rev. per min. of compressor	Cubic feet of piston displacement per minute	Volumetric efficiency	Temperature of intake air, deg. F.	Cubic feet of free air per minute	Gallons of water per minute pumped	Cubic feet of water per minute	Boiler pressures	Air pressures (gauge): Starting pressure of well	Air pressures (gauge): (Operating) from the indicator diagrams	Air pressures (gauge): (Operating) at receiver	Air pressures (gauge): (Operating) at well top	Static head in well	Total pumping head	W.H.P.	A.H.P.	Length of air line, feet	Submergence, feet	Submergence, per cent	Pumping efficiency
1	92	228.5	90.5	70	202.4	1100.0	147.2	100	153.0	141.0	139.0	138.0	3' 9"	37.0	10.3	43.9	357	321.3	90	26.5
2	87	216.0	93.0	70	197.0	1105.0	147.5	100	102.0	89.0	88.0	87.0	3' 10"	37.1	10.4	33.6	238	202.4	85	31.0
3	92	228.5	95.0	72	213.4	1108.0	147.6	105	76.0	63.5	62.0	61.5	3' 11"	37.1	10.4	29.8	179	143.2	80	35.0
4	104	258.7	96.0	74	240.6	1106.0	147.5	105	62.0	50.0	48.0	47.5	3' 10"	37.1	10.4	28.5	143	107.4	75	36.6
5	118	293.2	96.7	77	275.1	1100.0	147.2	100	50.5	37.5	36.5	36.0	3' 11"	37.0	10.3	27.4	119.3	83.4	70	37.7
6	140	349.2	96.7	83	321.7	1105.0	147.5	100	44.5	32.5	30.5	30.0	3' 10"	37.1	10.4	28.3	104.8	69.3	65	36.8
7	174	431.9	96.7	75	405.8	1107.0	147.5	100	37.5	24.5	23.6	23.2	3' 11"	37.1	10.4	30.2	89.5	53.7	60	34.5
8*	150 134	583.1	91.5†	78	517.0	1100.0	147.2	100	33.0	21.0	19.5	19.0	3' 10"	37.0	10.3	33.3	79.7	43.9	55	31.0
9*	180 210	775.2	93.0†	77	675.2	1100.0	147.2	105	29.5	17.5	16.0	15.5	3' 11"	37.1	10.3	39.0	71.6	35.8	50	26.5

* Both compressors used. † Average volumetric efficiency.

volume. Both compressors had to be operated at a considerably higher speed than their catalogue rating in order to test even at 50 per cent submergence.

As previously stated, it is usually desired to pump the well to its maximum economical capacity, but there are instances

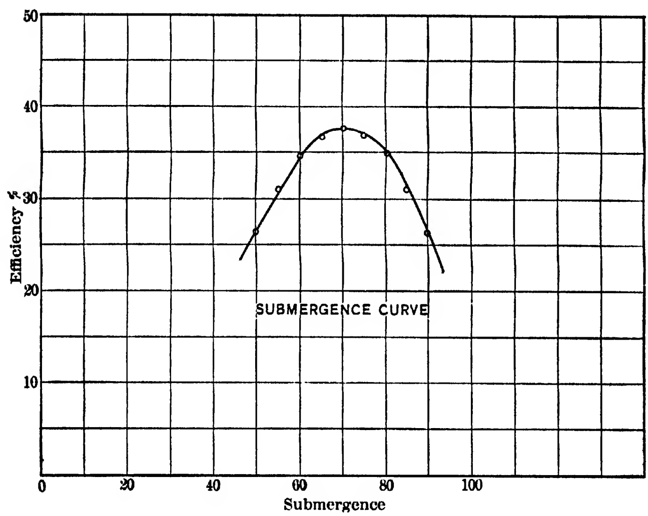


Fig. 60.

where a certain specified amount of water is desired from the well. In such cases, the first step necessary is to ascertain what the head drop in the well is when pumped to this required capacity. This may be done by choosing at random any submergence percentage and forcing sufficient air into the well to pump the required quantity. The drop in head will be shown on the air gauge. The air line is next varied in length and a series of tests made as just explained. To a certain extent, the method of piping the well for air affects the submergence curve. Some systems require for efficient operation greater submergence than others; the design and finish of air nozzles of any one system require more or less submergence.

No set rules can be given as to what is proper submergence and no formulæ can be derived that will be even an approximate guide. It is purely a matter of experiment in each individual case, and an air lift should never be installed without such experimenting. It is always best to obtain the advice of experienced men in any installation, for an air lift improperly designed and installed is one of the most criminally wasteful means of pumping known.

CHAPTER VI

VELOCITIES

It has been shown in preceding chapters that the greatest loss encountered in air-lift practice is that due to the slippage of air past the water in the ascending column. It has also been demonstrated how this loss varies with the different depths of submergence of the air piping. We come now to another factor which affects very materially the efficiency of operation, and that is the design of the discharge piping to be installed in the well. In selecting discharge pipe sizes or, in other words, in fixing the velocity of the column of mixed air and water, there can be no other guide but experiment and experience, and the designer is limited and handicapped at every turn by the sizes of standard pipe and the small area of the bored well.

To explain the difficulties and the necessity for experience, consider the conflicting demands of efficiency when transmitting a mixture of compressed air and water as is done by the air-lift. Referring to the laws of friction of water flow in a subsequent chapter, it is seen that the losses decrease as the pipe size is increased; referring to the air-lift theory in Chapter IV, it is seen that the air-slippage losses increase as the velocity of flow is diminished, or as the discharge pipe is increased. Then, for efficiency's sake, water demands a large pipe, and air, a small one.

There must be a point or, more correctly, a velocity of flow of mixed air and water, where the sum of the water-friction losses and the air-slippage losses is least, but owing to the variables involved, this velocity can only be determined by experiment.

The discussion so far refers to velocity at the point of admission of air into the discharge pipe. As the air bubbles ascend, pressure falls, as before explained, and, consequently, each as-

cending bubble expands steadily and occupies a steadily increasing space in the discharge pipe. This reduces the effective area for water flow, and, therefore, to maintain a constant quantity of water through the discharge pipe length, the velocity of the column must also increase as it ascends. The velocity of travel at any point in the pipe is expressed by the formula used in a subsequent chapter in hydraulic computations, or

$$Q = av \tag{56}$$

In the present instance, Q = cubic feet of air plus cubic feet of water per second, while a and v equals cross-sectional area of the pipe in square feet and the velocity in feet per second, respectively, as before. The air volume is the variable factor and it increases from bottom to top of discharge pipe very nearly in accordance with the following formula:*

$$v = Q_a\left(1 - \frac{x}{l}\left[1 - \frac{1}{r}\right]\right) \tag{57}$$

v = volume of compressed air in cubic feet per second;

Q_a = air volume in cubic feet of free air per second delivered by the compressor;

x = distance from discharge pipe top to the point where the volume is to be determined,

l = total length of the discharge pipe;

r = ratio of air compression.

The author has no knowledge of any reliable experimental data obtained from tests made under working conditions, and that were intended to show just what the most advantageous velocities should be in a discharge pipe. It can be said, however, that the velocity of mixed air and water should at no point be as low as the velocity with which air will ascend when submerged in still water, for then operation would be impracticable; on the other hand, the velocity at no point should be so high that the

* From *Compressed Air*, by Prof. Elmo G. Harris.

water-friction losses will overcome any gain obtained by reduction of the air-slippage losses. The velocity then is fixed between two rather indefinite limits, and all that remains to be said is, that as the ascending air volume increases, the column velocity should increase also and thereby minimize the sum of the air-slippage and water-friction losses.

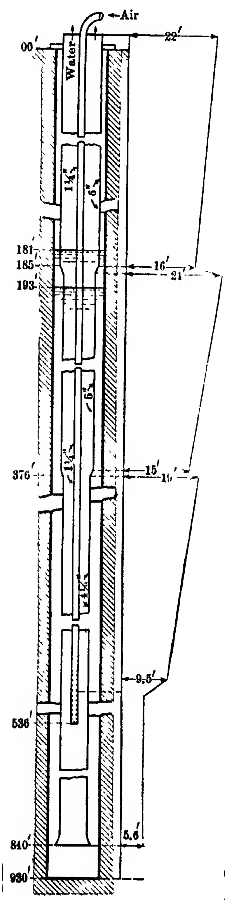


Fig. 61.

In a discharge pipe of uniform diameter from bottom to top, the velocity increases as the column ascends, thereby meeting the requirements just mentioned. In long discharge pipes of uniform area, the velocity of flow becomes excessive as the top is approached, hence it would appear advantageous to gradually increase the pipe diameter as it ascends. Some authorities seem to think that a column velocity of from six to twelve feet per second at the bottom, and from eighteen to twenty-five feet per second at the top is productive of good efficiency. Prof. Elmo G. Harris in *Compressed Air* states that an efficiency of 45 per cent was obtained in a well at Rolla, Mo., which was so piped that an initial velocity of 9.5 feet and a discharge velocity of 22 feet per second were obtained. The well and pipe details are shown in Fig. 61. This is indeed a remarkably high efficiency, the lift considered, and, in fact, is considerably higher than any the writer has found under similar conditions.

The author had occasion to test the air-lift plant, shown in Fig. 62, of the Mississippi State Insane Hospital at Jackson, Miss. The velocity at the lower end of the discharge pipe was 11.4 feet

per second, and at point of discharge was 8.5 feet per second. The mean results of this test were as follows:

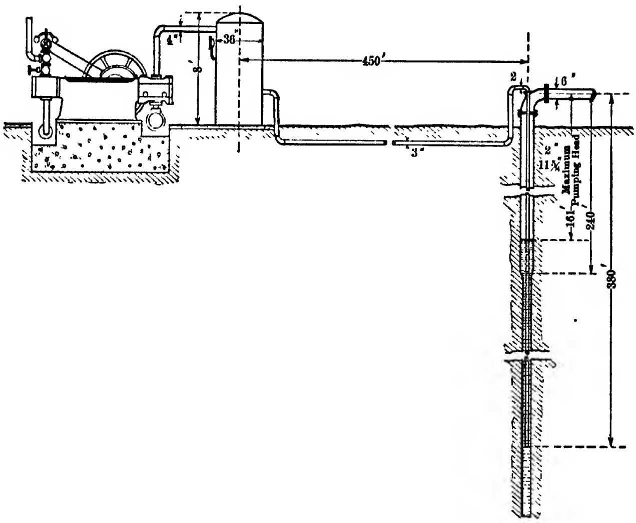


FIG. 62.

Compressor size.	10″ and 16″ × 12″
R.p.m.	165
Actual cubic feet of free air per minute	316
Starting pressure (gauge) at well, pounds. . .	110
Operating pressure (gauge) at well, pounds. .	100
Static head in well, feet.	135
Total pumping head, feet.	161
Gallons of water per minute.	441
A.H.P.	48
A.H.P. (isothermal)	42
W.H.P.	18
Pumping efficiency $\frac{\text{W.H.P.}}{\text{A.H.P. (isothermal)}}$	43 per cent
Over-all efficiency of plant.	30 per cent

These results show that good efficiency can be obtained when the dischaige velocity is lower than the initial velocity. There is no doubt, however, that considerable slippage losses occurred

in this installation and particularly at the point of increase in size of casing. The results of a test, made with an 8-inch discharge pipe extending from the swadge nipple to the point of overflow, would have been interesting.

In designing well piping practical limitations often make it impossible to exercise sufficient care. The bore of the well is usually small, hence the diameter of the pipe it will accommodate is very much limited. Then, too, standard pipe sizes prohibit nicety of design even were well diameters sufficiently large. The air line should be as large as possible for obvious reasons. An air velocity of 30 feet per second is considered good practice, but lower velocities should be employed when possible.

Practical Design. — With the formulæ, tables and other information given in this and preceding chapters, it is a simple matter to prepare designs that will be safe in practice and within 5 to 10 per cent of the highest obtainable efficiency. Best efficiency may be obtained only by running tests and making changes that the results indicate after the plant has been erected.

The data necessary for intelligent recommendations are as follows:

1. Number of wells to be pumped;
2. Entire depth of each;
3. Inside diameter of casing at top and bottom and, if diameter is reduced, to what depth is reduction made, and to what diameter;
4. Location, length and diameter of strainer, if used;
5. If no strainers are used, to what depth well is cased;
6. Gallons of water per minute to be pumped;
7. Static and pumping heads;
8. Distance from contemplated location of compressor to wells;
9. Horizontal and vertical distances that water is to be conveyed after leaving mouth of well;
10. Type of compressor preferred.

To illustrate the steps in designing an air-lift plant, assume a set of conditions as follows:

Depth of well, feet..................	500
Diameter of well, inches, no reductions.	8
Diameter of strainer, inches..........	7¾
Length, feet.......................	40
Located below ground surface, feet....	450
Gallons of water per minute.........	250
Static head, feet.....................	50
Pumping head, feet.................	65
Vertical lift above ground, feet.......	10
Horizontal distance, feet...........	200
Location of compressor from well, feet.	500
Type of compressor....	...Straight-line, steam-driven

The total lift, neglecting friction, is 75 feet. The first thing to be computed is the air pressure and, to do this, submergence must be fixed or assumed. Under conditions as stated about 65 per cent submergence is a good selection. The submergence in feet will be $1.5 \times 75 = 112.5$ and the operating pressure, neglecting the slight reduction caused by $\left(\frac{V^2}{2g} \times 0.434\right)$, will be $112.5 \times 0.434 = 48.8$ pounds. The length of air line from well top to lower end is $112.5 + 65 = 177.5$ feet and the discharge line, $112.5 + 75 = 187.5$ feet. To the bottom of the discharge line should be added about twenty feet of straight pipe having a bell mouth to reduce the entrance losses of the water.

The number of cubic feet of free air necessary is next to be ascertained. This may be computed by substitution in one of several empirical formulas. A very good formula proposed in part by Mr. Ed. Rix and in part by Mr. T. H. Abrams is as follows:

$$V_a = \frac{h}{C \log \frac{H+34}{34}} \qquad (58)$$

where

V_a = cubic feet of free air per minute (piston displacement) per gallon of water;

h = head in feet;

C = constant;

H = submergence in feet.

The values of C for various lifts are given as follows:

10 to 60 feet inclusive	245
61 to 200 feet inclusive	233
201 to 500 feet inclusive	216
501 to 650 feet inclusive	185
651 to 750 feet inclusive	156

Substituting the values of the present assumed conditions and the calculations therefrom for the symbols in the formula we have:

$$V_a = \frac{75}{233 \log \dfrac{112.5 + 34}{34}}$$

and solving:

$$V_a = 0.51 \text{ cubic feet per gallon.}$$

The free air capacity per minute required for the well is $0.51 \times 250 = 127.5$.

Regarding air-pressure and free-air-capacity requirements, it is well to add in passing that a compressor of from 10 to 15 per cent reserve capacity in both be selected for future contingencies. This because at some not distant date more wells may be drilled in the same stratum by other parties in the neighborhood and lower the pumping head of the present well, or the pumping head may be lowered by geological causes, as often happens. This would necessitate changes in piping and consequent increase in air volume and pressure. An 8″ and 9″ × 12″ compressor operating at 160 revolutions per minute has a capacity of 142 cubic feet of free air per minute at 50 to 75 pounds pressure. This machine would fit the requirements very satisfactorily.

The surface air line, designed according to principles given in a succeeding chapter, will be 2 inches in diameter. This size will convey the air with a pressure drop of about 1.3 pounds between receiver and well top. Inside the well, a $1\frac{1}{2}$-inch line will convey the air at a velocity of about 29 feet per second with a pressure drop of about 1.5 pounds between well top and the lower end of the air line.

The water-discharge-pipe size may be determined by first

computing the volume of compressed air passing through the air nozzle per second (formula $PV = P_1V_1$ will apply) and adding to this the number of cubic feet of water per second pumped. Next assume an initial column velocity within the limits given, substitute in the formula and solve for a as follows:

$$Q = av$$
$$0.425 + 0.56 = a \times 9$$
$$a = 0.109 \text{ square feet} = 15.7 \text{ square inches.}$$

The 15.7 square inches is the net area required for mixed air and water travel. Since the air line is to be suspended inside the discharge pipe, the outside area of the former must be added to 15.7 square inches to give the total area of discharge pipe required. In Table 27, Chapter XII, the outside area of 1½-inch pipe is 2.835 square inches, hence the total area of discharge pipe at point of admission of air must be 15.7 + 2.835 = 18.535.

Again, referring to Table 27, it is seen that the area of a 5-inch pipe is 19.986 square inches, and of a 4½-inch pipe 15.961 square inches. Reducing these areas to square feet, substituting in the formula and solving this time for v, it is found that, by using a 4½-inch pipe, the initial column velocity would be about 10 feet per second and, by using a 5-inch pipe, the initial velocity would be about 8 feet per second. Both pipe sizes, then, afford initial velocities within the limits of what is considered good practice, and the question now is, which of the two sizes is the better for practical purposes.

The discharge velocity for a pipe of uniform diameter of 5 inches would be about 21 feet per second, while the discharge velocity of a 4½-inch pipe would be about 27 feet per second. Hence, a 4½ discharge should be increased at a point where the velocity is about 20 feet per second to 5 inches in order to lower the discharge velocity and likewise lower the water-friction loss. At the point where the discharge pipe is increased, there occur losses due to sudden expansion of section and also eddy losses, which cause additional air slippage; therefore, it is best where the initial and discharge velocities are within the required limits,

as they are here in 5-inch pipe, to use a discharge line of uniform diameter. In other words, unbroken velocity lines are preferable to broken ones.

The conditions in the installation in question demand that the water be raised 10 feet above ground and transmitted horizontally a distance of 200 feet. If a closed line were used between the well and the point of discharge the friction imposed would be excessive; for it will be remembered that the velocity of flow at well top is 21 feet per second. Besides this loss, when air and water are transmitted horizontally and especially after striking an obstruction, the air rises to the top of the pipe and rides over the water, consequently considerable energy is dissipated. This may be overcome by extending the pipe vertically from the well a certain distance in addition to the ten feet and there separating the air from the water. The friction loss then to be contended with is only that of the water. The additional height of pipe necessary is the sum of the water friction head and the head required to produce the flow of water. Using a 5-inch surface line, this amounts to, including four elbows,

$$
\begin{aligned}
h &= f\frac{l}{d}\frac{v^2}{2g} + 4\left(f_1\frac{l_1}{d}\frac{v^2}{2g}\right) + \frac{v^2}{2g} \qquad (59)\\
&= 3.01 + 0.924 + 0.3 = 4.23 \text{ feet}
\end{aligned}
$$

or say, for safety, 4 feet, 6 inches.

This additional 4 feet 6 inches to be pumped against reduces the submergence percentage to 58.6 per cent and increases the pumping head to 79.5 feet. Water-friction losses in the well have been disregarded as it is impossible even to estimate these intelligently. Fig. 63 shows the completed design.

As before stated, it is well to install a discharge line some 20 feet deeper than the air line; so that, when finally adjusting the submergence from tests, only the air line will have to be handled.

Very often on starting a well after a period of idleness, large quantities of sand are discharged with the water. To prevent this sand from clogging the horizontal line or being conveyed to the reservoir, the valves shown by A, B and C in Fig. 63 are in-

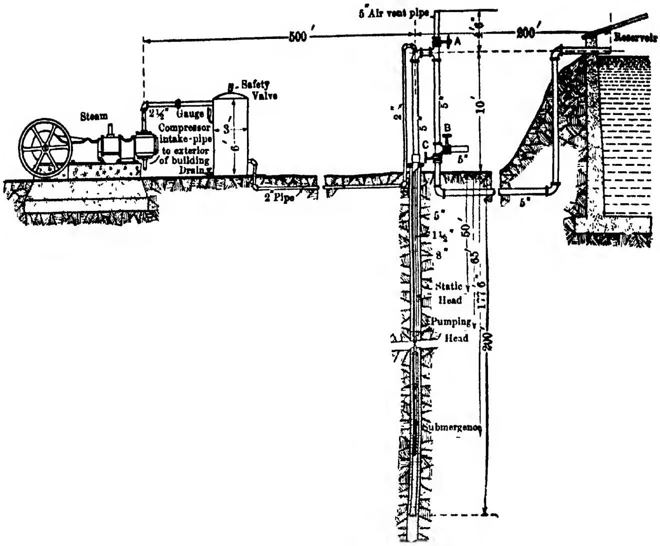

5" Air vent pipe
500'
200'
A
Reservoir
Safety Valve
Steam
2½" Gauge
Compressor intake-pipe to exterior of building
Drain
2" Pipe
B
C
10'
Static Head
Pumping Head
Submergence
200'

FIG. 63.

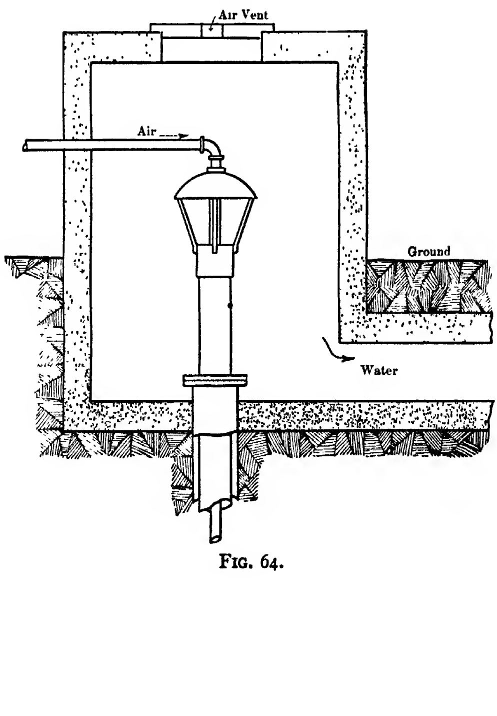

Air Vent
Air
Ground
Water

FIG. 64.

stalled. By closing C and A and opening B, the sand-laden water is allowed to go to waste, and after clearing, the water is permitted to flow to the reservoir by closing B and opening C and A.

At regular intervals the well should be washed out; otherwise sand will pack around the strainer, both inside and out, and restrict the water flow. Washing can be done very effectively by closing all three valves in the discharge

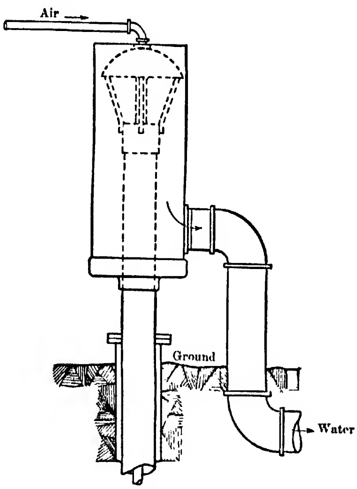


FIG. 65.

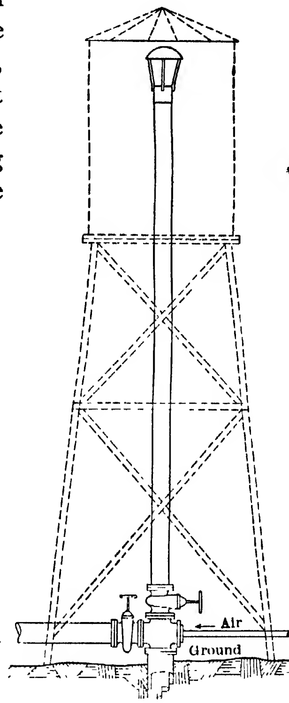


FIG. 66.

line and operating the compressor at maximum speed. When the highest allowable pressure is reached, suddenly open valve B, then, after the head has been blown off and the pressure reduced, close B and repeat the operation as long as sand appears with the water. When all surface outlets are closed and, as the air pressure builds up, the water is forced back through the strainer into the stratum; when the surface valve is opened, the water rushes back through the strainer into the well. The continuous outgoing and incoming rush of water will remove any

sand that may have found its way into the well, and also loosen and remove sand that may have become packed on the outside around the strainer walls.

Figures 64, 65 and 66 show various other methods employed for conveying water horizontally and vertically after leaving the well. In Fig. 66 the air-lift is shown discharging in a tank elevated some distance above the ground surface. This arrangement is seldom to be recommended because, as will be seen later, the efficiency of pumping falls off rapidly as the lift is increased. It is best to pump to the surface, or only a few feet above, with the air lift and then employ other and more efficient apparatus for further elevation.

CHAPTER VII

PUMPING SYSTEMS

CENTRAL PIPE (OPEN END) SYSTEM

A system of well piping very often found is that shown in Fig. 67. It consists, as shown, of a discharge line suspended from flange *C*, inside of which, suspended from an elbow, is the air line. Sometimes the discharge line is dispensed with and only the air line is suspended from the well top. In such installations the well casing serves as the discharge line. At best this system is a rough and ready affair with nothing but its simplicity to recommend it. Probably the inefficiency is largely accountable to the sudden change in direction of flow that the air must take to follow the water upward and out of the discharge pipe. The down coming air strikes the upward moving water column and undoubtedly, it is considerably retarded in its flow.

Performance. — A test on this system was run by the writer on a well owned by the Houston Ice & Brewing Co., at Houston, Texas. The methods of conducting the test, making observations, etc., were quite the same as previously described. The results were as follows:

Total depth of well, feet	610
Diameter of casing, inches	8
Diameter of discharge pipe, inches	3½
Diameter of air pipe, inches	1¼
Length of discharge pipe, feet	355
Length of air pipe, feet	320
Static head in well, feet	85
Distance above surface water lifted, feet	9
Total pumping head, feet	115
Starting pressure at well, pounds	107
Operating pressure at well, pounds	94
Gallons of water per minute pumped	120
Actual cubic feet free air per minute	74.5
Submergence, per cent	65
A.H.P. by indicator diagrams	12.1
A.H.P. (isothermal)	9.5
W.H.P	3.6
Pumping efficiency, per cent	37
Over-all efficiency, per cent	29.7

From data obtained from a number of tests of this system of piping made by the author in different localities and operating under varying conditions of lift and yields, the curves shown in Fig. 68 have been plotted. The upper, or submergence, curve is the average line drawn through the points of most advantageous percentages of submergence found at various pumping heads. The lower, or efficiency, curve is also the average line drawn through efficiency points corresponding to the submergence percentages. The curves give an excellent idea of what may be expected from the system under all conditions of lift up to 650 feet and also show the submergences necessary to best economy at each lift. They will be found quite useful in design and installation, but must not be taken as absolutely accurate and final, for they are intended to illustrate average results obtained from a number of tests, and local conditions in any specific instance may necessitate deviations.

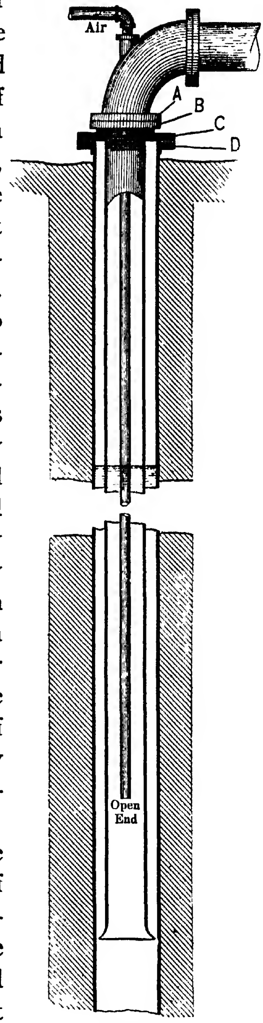


FIG. 67.

Table 12 was compiled from these curves. A table of this sort is of considerable more value than approximate empirical formulæ, because the values are based on practical trials and not dependent upon constants that usually embody a much too high factor of safety. The free air volumes are actual and, in computing compressor sizes, corrections must be made for volumetric effi-

TABLE 12
Central Pipe (Open End) System

Free lift	Per cent submergence	Air pressure	Length of air line, feet	Gallons of water per minute: 25 — Cubic feet of free air per minute	25 — Isothermal H.P.	50 — Cubic feet of free air per minute	50 — Isothermal H.P.	75 — Cubic feet of free air per minute	75 — Isothermal H.P.	100 — Cubic feet of free air per minute	100 — Isothermal H.P.	150 — Cubic feet of free air per minute	150 — Isothermal H.P.	200 — Cubic feet of free air per minute	200 — Isothermal H.P.	250 — Cubic feet of free air per minute	250 — Isothermal H.P.	300 — Cubic feet of free air per minute	300 — Isothermal H.P.	350 — Cubic feet of free air per minute	350 — Isothermal H.P.	400 — Cubic feet of free air per minute	400 — Isothermal H.P.	500 — Cubic feet of free air per minute	500 — Isothermal H.P.
25	79	41	119	3.3	0.28	6.6	0.6	10	0.8	13	1.1	20	1.6	26	2.2	33	2.8	40	3.4	46	3.9	52	4.5	66	5.6
50	76	68	208	5.9	0.65	12	1.3	18	1.9	24	2.6	35	3.9	47	5.2	59	6.5	71	7.8	83	9.1	94	10.4	118	13.0
75	72	84	268	8.8	1.08	18	2.2	26	3.2	35	4.3	53	6.4	70	8.6	88	10.8	106	12.9	126	15.4	140	17.2	176	21.6
100	68	93	313	12.2	1.56	24	3.1	37	4.7	49	6.2	73	9.4	98	12.5	122	15.6	146	18.7	164	21.7	196	25.0	244	31.2
125	65	100	358	15.8	2.08	32	4.2	47	6.2	63	8.3	95	12.5	126	16.6	158	20.8	190	25.0	224	29.4	252	33.2	316	41.6
150	63	110	407	19.8	2.71	40	5.4	59	8.1	79	10.8	119	16.3	156	21.7	198	27.1	238	32.5	280	37.8	312	43.4	396	54.2
175	61	119	450	23.6	3.35	47	6.7	71	10.1	94	13.4	142	20.1	189	26.8	236	33.5	284	40.2	329	46.9	378	53.6	472	67.0
200	59	126	488	27.8	4.02	55	8.0	83	12.1	111	16.1	167	24.1	222	32.2	278	40.2	334	48.2	385	56.0	444	64.4	556	80.4
250	56	138	568	37.0	5.55	74	11.1	111	16.7	148	22.2	222	33.3	296	44.4	370	55.5	444	66.6	518	77.7	592	88.8	740	111.0
300	53	147	639	47.7	7.33	95	14.7	143	22.0	191	29.3	286	44.0	383	58.6	477	73.3	572	87.9	665	102.9	...	...	...	...
350	50	151	700	59.4	9.22	119	18.4	179	27.7	237	30.9	356	55.3	475	73.8	594	92.2	712	110.6	833	128.8	...	...	...	...
400	48	160	770	73.0	11.50	146	23.0	219	34.5	292	46.0	438	69.0	584	92.0	730	115.0	...	...	...	...	...	...	...	...
600	43	195	1050	115.5	19.70	231	39.4	347	59.1	462	78.8	693	118.2	924	157.6	1155	197.0	...	...	...	...	...	...	...	...

ciency and leakage. Also, the air pressures given must be corrected for frictional losses.

The Saunders' System. — This system has been discussed in a previous chapter. A practical installation is shown in Fig. 69.

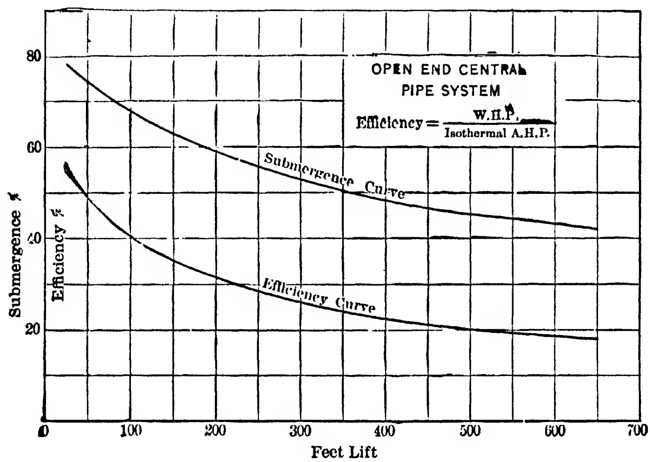


Fig. 68.

The discharge line is suspended from the flange *C* and the compressed air is admitted by means of the special fittings attached to the well top. One of the main objections to the Saunders' system is the likelihood of air leakage through the well casing. To successfully operate the system in a well having a defective casing, it is necessary to install an auxiliary pipe line inside the casing, and admit the air between this line and the discharge line.

Performance. — A test of the Saunders' system was made by the writer on oil well No. 32 of the Crowley Oil & Mineral Co., at Evangeline, La., and the results published in the *Proceedings of the American Society of Mechanical Engineers*, Vol. 31, page 311. A summary of the results is as follows:

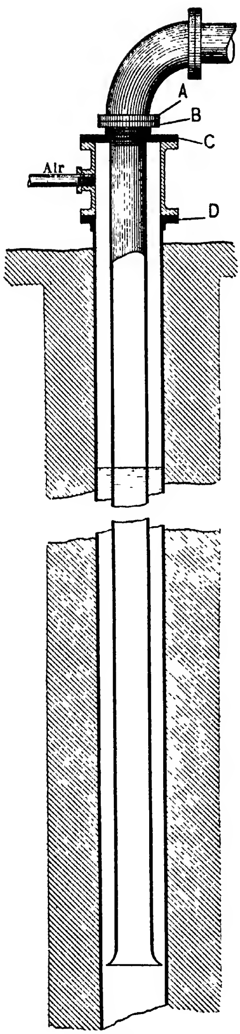


FIG. 69.

Total depth of well, feet	1805
Diameter of casing, inches	6
Diameter of discharge pipe, inches	4
Length of discharge pipe, feet	1513
Total pumping head, feet	1135.5
Operating air pressure at well, pounds	153
Gallons of fluids pumped per minute	32
Weight of one gallon of fluid, pounds	8.7
Percentage of salt water in fluid	87.3
Percentage of sand in fluid	2.2
Percentage of crude oil in fluid	10.5
Specific gravity of oil	0.9
Actual cubic feet of free air per minute	573.5
Percentage of submergence	25.0
A.H.P. by indicator diagrams	107.5
A.H.P. (isothermal)	89.5
W.H.P.	9 97
Pumping efficiency, % $\frac{\text{W.H.P.}}{\text{A.H.P. (isothermal)}}$	11.2
Over-all efficiency, %	9.3

Figure 70 is the submergence and efficiency curves plotted as before explained. Table 13 was computed from the curves in the manner explained.

The Pohle System. — Explanation of the principle and action of this system has already been given. Fig. 72 shows a practical installation and *C*, *D*, and *E* of the same figure are various types of foot pieces sometimes employed. The writer has never tested either types *D* or *E*. The results following were obtained from tests on the other two types shown.

Performance. — A test of the Pohlé system of type shown at *C*, Fig. 71, was made by the writer in one of two wells owned by the Armstrong Packing Co., at Dallas, Texas. The methods employed were identical with those before mentioned. A summary of results is as follows:

TABLE 13
Saunders' System

Feet lift	Per cent submergence	Air pressure	Length of air line, feet	25 gal.: Cubic feet of free air per minute	25: Isothermal H.P.	50: Cubic feet of free air per minute	50: Isothermal H.P.	75: Cubic feet of free air per minute	75: Isothermal H.P.	100: Cubic feet of free air per minute	100: Isothermal H.P.	150: Cubic feet of free air per minute	150: Isothermal H.P.	200: Cubic feet of free air per minute	200: Isothermal H.P.	250: Cubic feet of free air per minute	250: Isothermal H.P.	300: Cubic feet of free air per minute	300: Isothermal H.P.	350: Cubic feet of free air per minute	350: Isothermal H.P.	400: Cubic feet of free air per minute	400: Isothermal H.P.	500: Cubic feet of free air per minute	500: Isothermal H.P.
25	78	39	114	3.4	0.28	7	0.6	10	0.8	14	1.1	20	1.6	27	2.2	34	2.8	41	3.4	48	3.9	54	4.5	68	5.6
50	74	61	192	5.8	0.61	12	1.2	17	1.8	23	2.4	35	3.6	46	4.8	58	6.1	70	7.2	81	8.6	92	9.6	116	12.2
75	70	78	250	8.6	1.01	17	2.0	26	3.0	34	4.0	51	6.0	69	8.0	86	10.1	103	12.0	120	14.1	138	16.0	172	20.2
100	67	89	320	12.4	1.44	26	2.9	37	4.3	50	5.8	74	8.6	99	11.6	124	14.4	149	17.2	182	20.2	198	23.2	248	28.8
125	65	102	358	14.5	1.93	29	3.9	44	5.8	58	7.8	87	11.6	116	15.6	145	19.3	174	23.2	203	27.3	232	31.2	290	38.6
150	63	110	407	18.3	2.50	37	5.0	55	7.5	73	10.0	110	15.0	146	20.0	183	25.0	220	30.0	259	35.0	292	40.0	366	50.0
175	60	114	438	22.5	3.13	45	6.3	68	9.4	90	12.6	135	18.8	180	25.2	225	31.3	270	37.6	315	44.1	360	50.4	450	62.6
200	58	120	478	26.3	3.73	53	7.5	79	11.2	105	15.0	158	22.4	210	30.0	263	37.3	316	44.8	371	52.5	420	60.0	526	74.6
250	55	132	556	34.5	5.10	69	10.2	104	15.3	138	20.4	207	30.6	276	40.8	345	51.0	414	61.2	483	71.4	552	81.6	690	102.0
300	52	140	625	43.5	6.56	87	13.1	131	19.7	174	26.2	261	39.4	348	52.4	435	65.6	522	78.8	609	91.7	...	...	...	...
350	50	152	700	52.7	8.20	105	16.4	158	24.6	211	32.8	316	49.2	422	65.6	527	82.0	632	98.4	735	114.8	...	...	...	...
400	49	167	785	62.1	9.95	124	19.9	186	29.9	248	39.8	373	59.8	497	79.6	621	99.5	...	...	...	...	...	...	...	...
500	45	178	910	80.1	13.20	160	26.4	240	39.6	320	52.8	481	79.2	641	105.6	801	132.0	...	...	...	...	...	...	...	...
600	42	186	1032	103.0	17.30	206	34.6	309	51.9	412	69.2	618	103.8	...	...	...	...	...	...	...	...	...	...	...	...
800	37	204	1269	162.7	28.10	325	56.2	488	84.3	651	112.4	976	168.6	...	...	...	...	...	...	...	...	...	...	...	...
1000	31	195	1461	231.5	39.50	463	79.0	695	118.5					...	...	...	...	...	...	...	...	...	...	...	...
1200	30	225	1715	280.4	50.71	561	101.4	841	152.1							...	...	...	...	...	...	...	...	...	...

Total depth of well, feet	729
Diameter of casing, inches	8
Diameter of discharge pipe, inches	$3\frac{1}{2}$
Diameter of air line, inches	$1\frac{1}{2}$
Length of discharge line, feet	350
Length of air line, feet	336
Total pumping head, feet	121
Starting pressure at well, pounds	102
Operating pressure at well, pounds	90
Gallons of water pumped per minute	135
Actual cubic feet of free air per minute	85
Submergence, per cent	64
A.H.P. by indicator diagrams	12.5
A.H.P. (isothermal)	10.7
W.H.P	4.1
Pumping efficiency, % $\frac{\text{W.H.P.}}{\text{A.H.P. (isothermal)}}$	37.5
Over-all efficiency, %	32.8

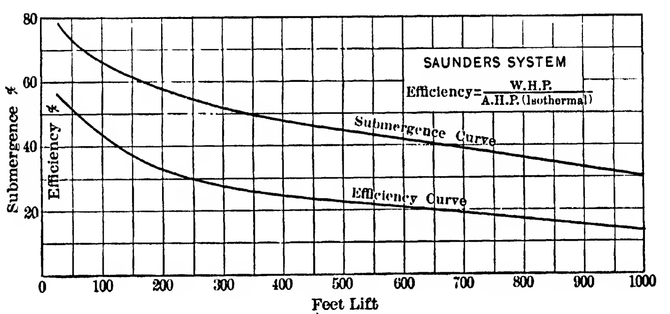


FIG. 70.

Figure 72 shows the curves plotted from a number of such tests and in Table 14 are the results computed from the curves.

Central Pipe (Perforated End) System. — Fig. 73 shows the usual installation of this system. The air and discharge piping is suspended as in the open-end system. Near the end of the air line, a number of holes one-eighth of an inch in diameter are drilled. The sum of the areas of the holes so drilled should be equal to at least one and one-half times the area of the air pipe. It is best to leave from 6 to 8 feet of blank pipe with lower end open,

TABLE 14
POHLE SYSTEM

Feet lift	Per cent submergence	Air pressure	Length of air line, feet	25: Cubic feet of free air per minute	25: Isothermal H.P.	50: Cubic feet of free air per minute	50: Isothermal H.P.	75: Cubic feet of free air per minute	75: Isothermal H.P.	100: Cubic feet of free air per minute	100: Isothermal H.P.	150: Cubic feet of free air per minute	150: Isothermal H.P.	200: Cubic feet of free air per minute	200: Isothermal H.P.	250: Cubic feet of free air per minute	250: Isothermal H.P.	300: Cubic feet of free air per minute	300: Isothermal H.P.	350: Cubic feet of free air per minute	350: Isothermal H.P.	400: Cubic feet of free air per minute	400: Isothermal H.P.	500: Cubic feet of free air per minute	500: Isothermal H.P.
25	77	36	109	3.5	0.28	7	0.6	11	.8	14	1.1	21	1.6	28	2.2	35	2.8	42	3.4	49	3.9	56	4.5	70	5.6
50	73	58	185	6.0	0.62	12	1.2	18	1.9	24	2.5	36	3.7	48	4.9	60	6.2	72	7.4	84	8.4	96	9.8	120	12.4
75	69	72	242	9.1	1.03	18	2.1	27	3.1	36	4.1	55	6.2	73	8.2	91	10.3	109	12.4	127	14.7	146	16.4	182	20.6
100	65	81	288	12.0	1.45	24	2.9	36	4.4	48	5.8	72	8.7	96	11.6	120	14.5	144	17.4	168	20.3	192	23.2	240	29.0
125	63	93	338	15.1	1.97	30	3.9	45	5.9	60	7.9	91	11.8	121	15.7	151	19.7	181	23.6	211	27.3	242	31.4	302	39.4
150	62	106	395	18.7	2.52	37	5.0	56	7.6	75	10.1	112	15.1	150	20.1	187	25.2	224	30.2	259	35.0	300	40.2	374	50.4
175	61	119	450	22.1	3.12	44	6.2	66	9.4	88	12.5	133	18.7	177	25.0	221	31.2	265	37.4	309	43.4	354	50.0	442	62.2
200	59	126	488	25.7	3.73	51	7.5	77	11.2	103	14.9	154	22.4	206	29.8	257	37.3	308	44.8	358	52.5	412	59.6	514	74.6
250	55	132	556	33.2	4.95	66	9.9	100	14.8	133	19.8	199	29.7	266	39.6	332	49.5	398	59.4	465	60.3	532	79.2	664	99.0
300	52	140	625	42.1	6.35	84	12.7	126	19.1	168	25.4	253	38.1	337	50.8	421	63.5	505	76.2	588	88.9	...	...	...	...
350	50	151	700	48.0	7.45	96	14.9	144	22.4	192	29.8	288	44.7	384	59.6	480	74.5	576	89.4	672	100.3	...	...	...	...
400	48	160	770	58.3	9.20	117	18.4	175	27.6	233	36.8	350	55.2	466	73.6	583	92.0	...	...	...	...	...	...	...	...
500	46	186	927	74.1	12.40	148	24.8	222	37.2	296	49.6	445	74.4	593	99.2	741	124.0	...	...	...	...	...	...	...	...

Gallons of water per minute: 25, 50, 75, 100, 150, 200, 250, 300, 350, 400, 500.

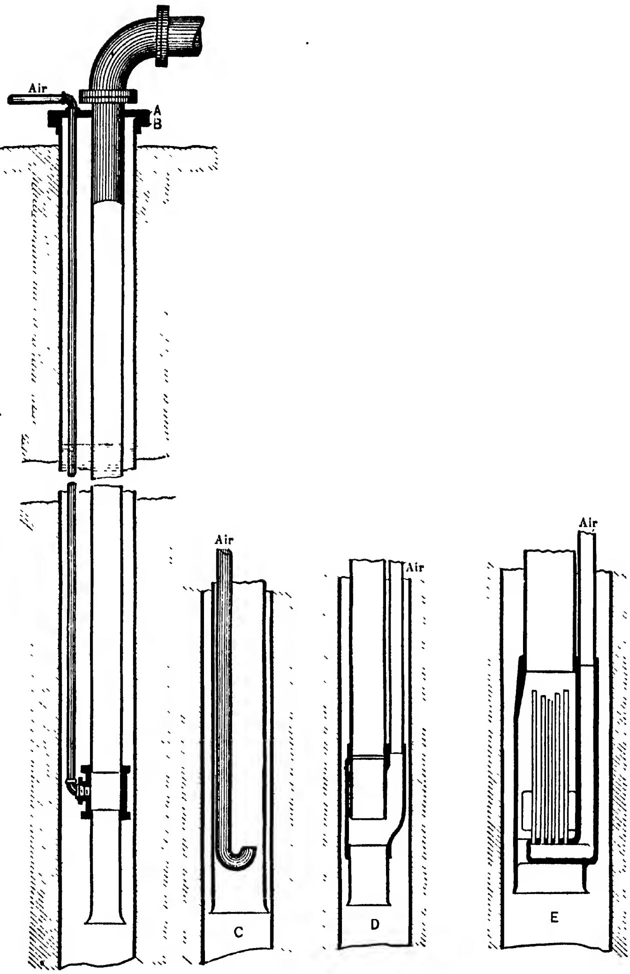


Fig. 71.

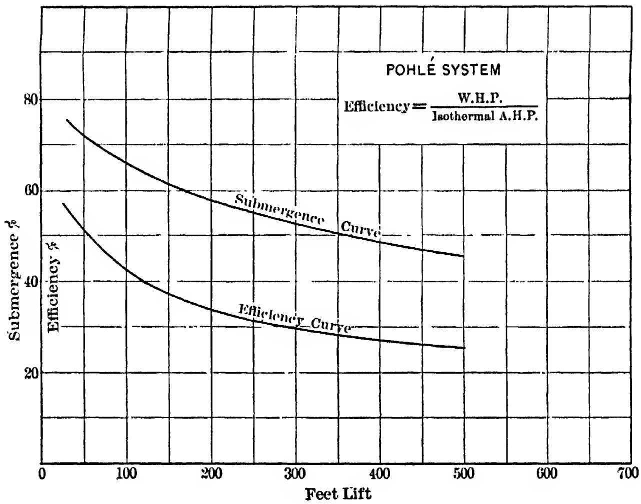


FIG. 72.

below the preforations; for if the lower end is plugged, scale and dirt from the air line will accumulate and will eventually clog the small openings. With the extension of the pipe no air will reach the pipe end, but instead, will follow the path of least resistance, which is through the holes. This is indicated in Fig. 73.

The small openings mentioned divide the air bulk into small streams and thoroughly aerate the column. Considerable gain in economy over other systems is realized thereby, as may be seen by comparison of the curves and tables.

Performance. — A test was made by the writer on one of two oil wells operated with this system and owned by the Mamou Power Co., of Evangeline, La. The methods employed in testing were identical with others mentioned, with the exception that the fluid pumped was measured by means of an 18-inch rectangular weir.

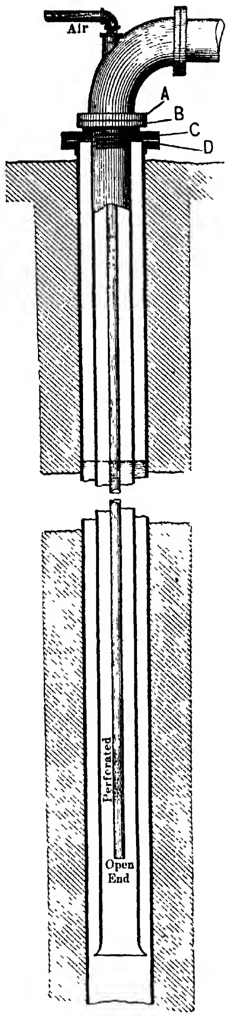


FIG. 73.

Total depth of well, feet	1900
Diameter of casing, inches	6
Diameter of discharge pipe, inches	4
Diameter of air line, inches	$1\frac{1}{4}$
Length of discharge pipe	1510
Length of air line	1492
Total pumping head	895
Operating air pressure at well, pounds	250
Gallons of fluid per minute pumped	45.2
Weight of one gallon of fluid, pounds	8.72
Percentage of salt water in fluid	87.7
Percentage of sand in fluid	1.2
Percentage of crude oil	11.1
Specific gravity of oil	0.9
Actual cubic feet of free air per minute	326
Percentage of submergence	39.8
A.H.P. by indicator diagrams	74.5
A.H.P. (isothermal)	60.7
W.H.P.	10.6
Pumping efficiency, % $= \frac{\text{W.H.P.}}{\text{A.H.P. (isothermal)}}$	17.5
Over-all efficiency, %	14.3

Figure 74 shows the submergence and efficiency curves plotted from such tests as this one, and in Table 15 are the results computed from the curves, as before.

Summary. — Fig. 75 shows all the efficiency curves plotted on one sheet. This gives an excellent comparison of the various systems, and the curves taken as a whole will be found very accurate statements of the possibilities of the air lift in-so-far as economy of operation is concerned. The curves show that the efficiency of all systems decreases with an increasing discharge head, other conditions remaining constant. It must be remembered that these curves are plotted from the results of tests of a large number of wells and every point represents the average results of a separate test.

TABLE 15
Central Pipe (Perforated End) System

Gallons of water per minute

Feet lift	Per cent submergence	Air pressure	Length of air line, feet	25: Cubic feet of free air per minute	25: Isothermal H.P.	50: Cubic feet of free air per minute	50: Isothermal H.P.	75: Cubic feet of free air per minute	75: Isothermal H.P.	100: Cubic feet of free air per minute	100: Isothermal H.P.	150: Cubic feet of free air per minute	150: Isothermal H.P.
25	77	36	109	3.4	0.27	7	0.5	10	0.8	14	1.1	20	1.6
50	72	56	179	5.8	0.59	12	1.2	17	1.8	24	2.3	34	3.5
75	70	78	250	8.1	0.95	16	1.9	24	2.9	32	3.8	48	5.7
100	65	81	286	11.4	1.37	23	2.7	34	4.1	46	5.5	68	8.2
125	63	93	338	14.4	1.84	29	3.7	43	5.5	58	7.3	86	11.0
150	60	97	376	18.3	2.38	37	4.8	55	7.1	74	9.5	110	14.3
175	59	108	427	21.2	2.88	42	5.8	64	8.6	84	11.5	128	17.3
200	57	115	466	24.6	3.42	49	6.8	74	10.3	98	13.7	148	20.5
250	53	123	533	31.6	4.53	63	9.1	95	13.6	126	18.1	190	27.2
300	50	130	600	39.6	5.85	79	11.7	119	17.6	158	23.4	238	35.1
350	47	139	661	47.4	7.15	95	14.3	142	21.5	190	28.6	284	42.9
400	45	143	728	55.5	8.45	111	16.9	167	25.4	222	23.8	334	50.7
500	43	164	878	71.0	11.27	142	22.5	213	33.8	284	45.1	426	67.6
600	42	186	1032	91.0	15.20	182	30.4	273	45.6	364	60.8	546	91.2
800	38	212	1289	131.5	23.00	263	46.0	395	69.0	526	92.0	790	138.0
1000	36	245	1562	190.0	35.10	380	70.2	570	105.3	...			
1200	33	256	1795	254.0	47.50	508	95.0	762	142.5				

Feet lift	200: Cubic feet of free air per minute	200: Isothermal H.P.	250: Cubic feet of free air per minute	250: Isothermal H.P.	300: Cubic feet of free air per minute	300: Isothermal H.P.	350: Cubic feet of free air per minute	350: Isothermal H.P.	400: Cubic feet of free air per minute	400: Isothermal H.P.	500: Cubic feet of free air per minute	500: Isothermal H.P.
25	28	2.2	34	2.7	40	3.2	49	3.8	56	4.3	68	5.4
50	48	4.7	58	5.8	68	7.0	84	8.2	96	9.4	116	11.7
75	64	7.6	81	9.5	96	11.4	112	13.3	128	15.2	162	19.0
100	92	10.9	114	13.7	136	16.4	161	19.2	184	21.9	228	27.4
125	116	14.7	144	18.4	172	22.1	203	25.8	232	29.4	288	36.8
150	148	19.0	183	23.8	220	28.6	259	33.3	296	38.1	366	47.6
175	168	23.0	212	28.8	256	34.6	294	40.3	336	46.1	424	57.6
200	196	27.4	246	34.2	296	41.1	343	47.9	392	54.8	492	68.4
250	252	36.2	316	45.3	380	54.4	431	63.4	504	72.4	632	90.6
300	316	46.8	396	58.5	476	70.2	553	81.9	.	..	...	...
350	380	57.2	474	71.5	568	85.8	665	100.1		.	....	...
400	444	67.6	555	84.5						...	...	...
500	568	90.2	710	112.7						..	....	...
600									...	..	....	...
800									...	...	....	...
1000								...	....	...	....	...
1200									....	...	....	...

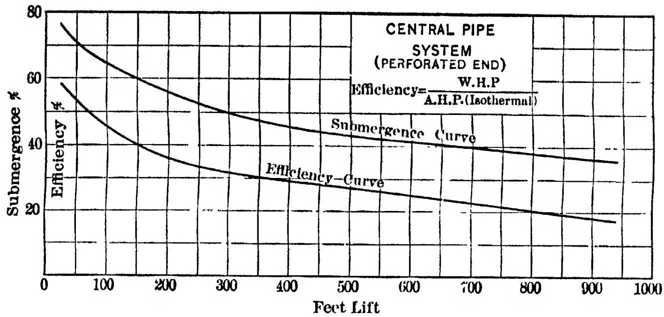

CENTRAL PIPE SYSTEM (PERFORATED END)
Efficiency= W.H.P / A.H.P.(Isothermal)
Submergence Curve
Efficiency-Curve
Submergence %
Efficiency %
80
60
40
20
0
100
200
300
400
500
600
700
800
900
1000
Feet Lift

FIG. 74.

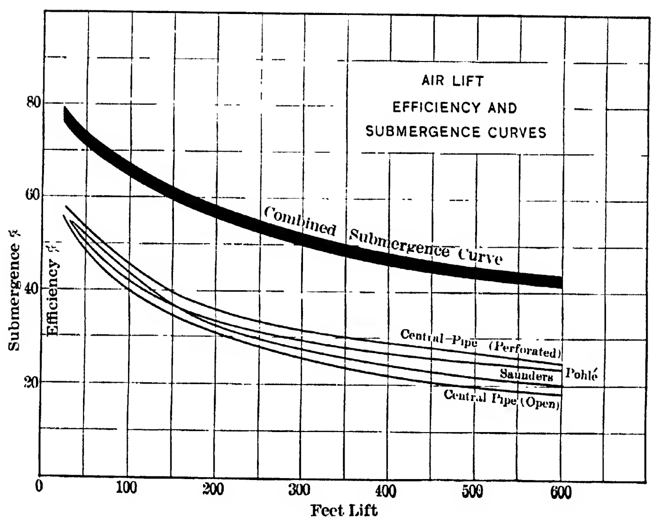

AIR LIFT EFFICIENCY AND SUBMERGENCE CURVES
Combined Submergence Curve
Central-Pipe (Perforated)
Saunders
Pohlé
Central Pipe (Open)
Submergence %
Efficiency %
80
60
40
20
0
100
200
300
400
500
600
Feet Lift

FIG. 75.

The efficiency curve plotted from tests in any particular well will be quite different. In nearly all the wells tested by the author, the efficiency increased with the lift up to a certain point, and further lift increase was accompanied by efficiency decrease. This is clearly indicated by the results of the test made on the well at Hattiesburg, Miss., given in Chapter V.

Special Applications. — The air lift is very often employed in the lifting and transmitting of many fluids and semi-fluids

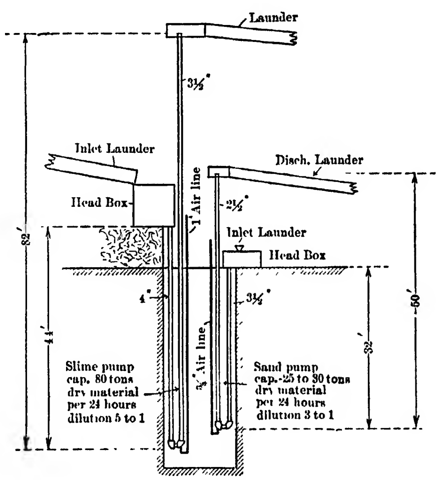


FIG. 76.

other than water. In a mine in Mexico, sand mixed with a small percentage of water is pumped with compressed air. The Colorado School of Mines Magazine describes this outfit in connection with the diagram reproduced in Fig. 76. The arrangement of piping, lift and other details are fully shown in the figure. The volume of air necessary to do the work was from $2\frac{1}{2}$ to 3 cubic feet of free air to each cubic foot of sand or slime, and the air pressure necessary was 28 pounds.

In Railway and Locomotive Engineering, a device for transferring grain with compressed air is described. Fig. 77 shows the

arrangement in detail. The lower air nozzles lift the grain to the elbow at the top, and the horizontal nozzle at that point furnishes a blast that transmits the grain horizontally to the desired place. It is said that the device, when supplied with air at 90 to 100 pounds, will do the work of five or six men.

Three combination patents on systems of mining sulphur have been granted to Mr. Herman Frasch. All systems consist

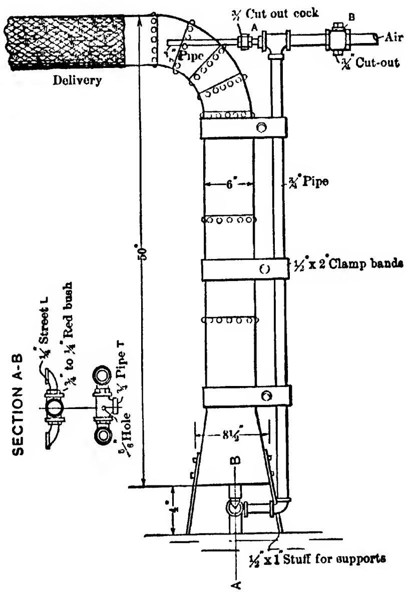


FIG. 77.

of a bored well penetrating the sulphur, closed surface discharge tanks and some means of pumping. In patent No. 461,429 the pump used is the familiar vertical direct-acting type with the plunger attached to rods and operating in a working barrel in the well; and in patent No. 461,430, a horizontal, hot-water steam pump is used to force the liquifying agent into the well and out of the discharge line.

The operation consists in liquefying the sulphur by either melting or dissolving and pumping the mixture to the surface into closed tanks, where the sulphur is settled. The hot water is drawn off the top and goes through the heaters again and thence to the well, again liquefying the sulphur, and so on.

For lifting the fluid compressed air has been found more satisfactory than either a horizontal or a vertical plunger pump. In Fig. 78 is shown the air-lift applied to this service.

The well casing extends a short distance into the sulphur and is there anchored. The uncased hole is continued down to the bottom of the stratum of sulphur and, next, the well is piped, as shown. Steam is admitted between the discharge line and well casing, and in a short time the sulphur is melted by it. The liquid sulphur fills the hole and rises up in the discharge pipe when air is admitted, and the mixture of molten sulphur and steam is lifted to the surface where the two are separated. Besides melting the sulphur, the steam also assists in the actual pumping.

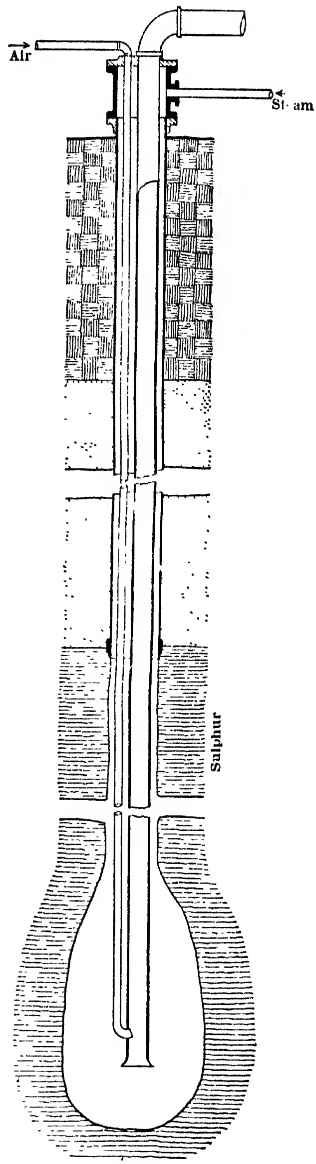


FIG. 78.

The air-lift has been applied successfully to the elevating of pulp in cyanide plants. The greatest difficulty to be overcome in this work is the disposal of obstructions, such as pieces of belting and chips which accumulate in the pipes.

An air-lift that may be quickly dismantled and is easily accessible is illustrated in Fig. 79. By removing the plug in the lower end of the tee, the standpipe may be drained and the air pipe may be taken out by disconnecting the union and removing the other plug. After cleaning, all parts may be easily and quickly replaced. This construction requires that the air-lift be located above the floor level and in place having sufficient clearance to permit dismantling.

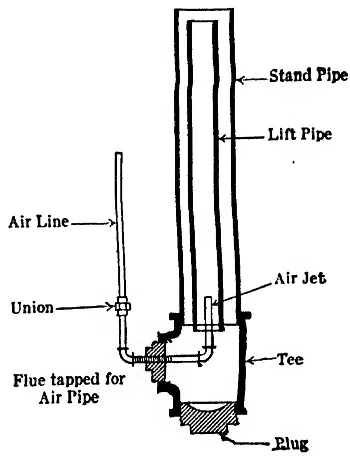


Fig. 79.

Fig. 80 illustrates a method of emptying oil barrels and at the same time elevating the contents into the storage tanks. The distance or the height of the storage tanks is immaterial so long as the head to be pumped against does not exceed the available air pressure. *A* is placed in the bung hole and a close fit is made possible by means of the taper sleeve shown at *e*. *C* is a sliding sleeve attached to the lower end of the nipple *a* which permits of the emptying of the entire contents of barrels of slightly varying diameters. *B* is a safety valve made from an old globe valve and shown attached at *f* or *A*. Air is admitted through the supply pipe shown and a pressure is exerted on the surface of the oil in the barrel. This forces the oil out through the nipple *a* and to the storage.

In *Coal Age*, Mr. S. W. Symons describes a method of draining mine workings with an air-lift that was employed on the holdings of the Clinton Coal Co., of Clinton, Ind. Water had

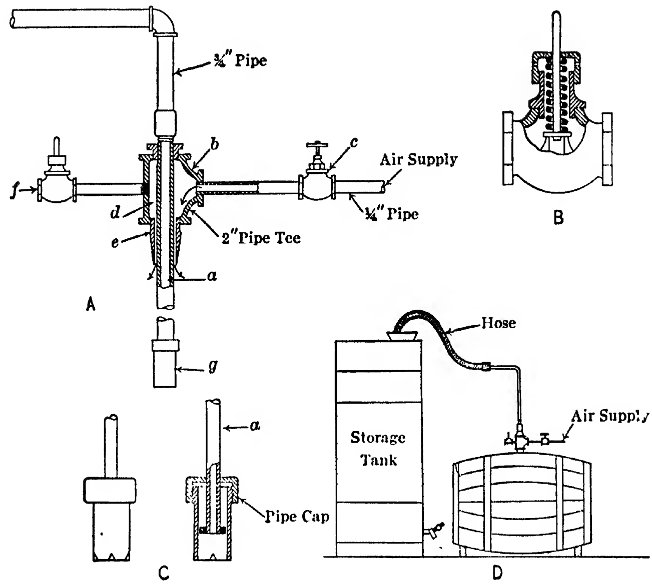


Fig. 80.

broken into the mine some 4000 feet from the shaft and about 160 feet from the surface. An air-lift was installed as clearly shown in Fig. 81 and after four months continuous operation, the water was lowered sufficiently to resume mining.

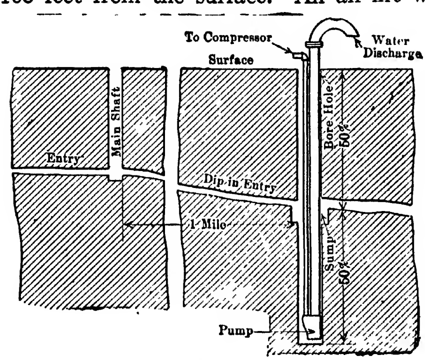


Fig. 81.

It is not always possible, however, to drill the bore-hole below the level of the workings which is obviously necessary to provide sufficient submergence for the air-lift. In such cases it may be found feasible to install either the compound lift described on page 113 or a displacement pump.

CHAPTER VIII

COMMERCIAL SYSTEMS

Several companies specialize in the manufacture of air-lift pumps for which are made more or less elaborate claims of superior economy. Tests comparing performances with one or more of the systems described in Chapter VII are often published to substantiate the claims. These manufactured systems consist of specially-designed head and foot pieces and in some, claims of superior efficiency are based upon refined designs of nozzles and deflector tubes properly placed in the foot piece. Whether or not any material gain can be so obtained has been the cause of considerable discussion and in a paper entitled *An Investigation of the Air Lift Pump*, by Professors Davis and Weidner, it is stated, after considerable experimenting, that "The type of foot piece has very little effect on the efficiency of the pumps, so long as the air is introduced in an efficient manner and the full cross-sectional area of the eduction pipe is realized for the passage of the liquid. Anything in the shape of a nozzle to increase the kinetic energy of the air is detrimental." The experiments on which this conclusion was based were made in the laboratory of the University of Wisconsin, and the well was constructed above the ground surface. Every possible

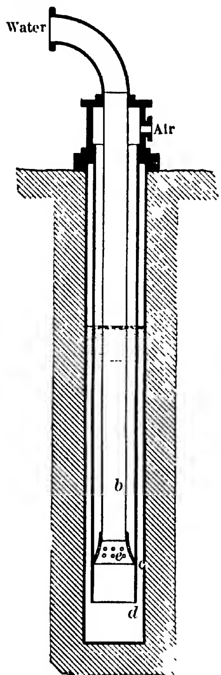


FIG. 82.

arrangement was made to obtain well conditions such as are met within the field, but, obviously, this was impossible. No head drop was provided, the lifts were low and the quantity of water pumped very small. While such experiments are always interesting, it is hardly fair to condemn an appliance when tested under conditions that were not contemplated when the design was prepared.

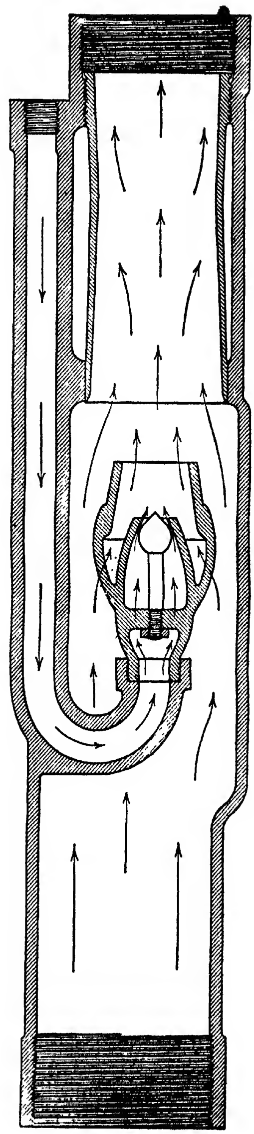


Fig. 83.

The Bacon System. — In 1895, James E. Bacon was granted a patent (No. 542,620) on an air-lift system, which consisted of a discharge pipe installed as in the Saunders' system, and provided near the lower end with a hole, thus admitting air to the water column before the surface of the water outside the pipe had been lowered to the end. The Hudson Engineering & Contracting Co. now manufacture an improved Bacon System, the details of which are shown in Fig. 82. Air is forced between the discharge pipe and either the well casing or the auxiliary pipe, which latter is shown installed. The water level is lowered until the holes in the foot piece are exposed, when, immediately, air is admitted to the water column, and operation begins.

The holes divide the air volume into a number of fine streams, and thorough aeration results. The bell mouth of the foot piece has for its ob-

ject the reduction of entrance losses of the water. No air pipe is employed in the well, and consequently, the friction losses of air in this pipe are eliminated and the full area of discharge pipe is available for water travel.

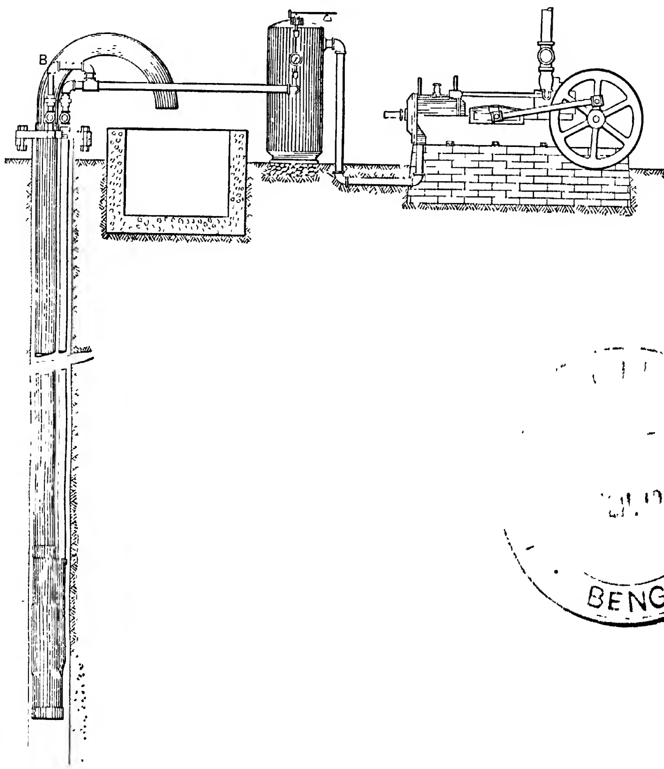


FIG. 84.

The Harris System.—In 1904, W. B. Harris was granted a patent (No. 814,601) on an air pump, which consisted of an ejector arrangement and contracted passageway. The pump, somewhat improved over the original design, is manufactured

by the Harris Air Pump Co., and is shown in section in Fig. 83. The arrows indicate the air and water travel and illustrate the aerating process. By enlarging the body of the pump around the air tube, uniform area for the travel of the water is obtained.

Figure 84 shows a Harris pump installed on the end of the discharge line in a well. The well-top connections are also shown. The branch pipe *B* tapped into the main line admits air between the well casing and discharge pipe, thus putting an outside

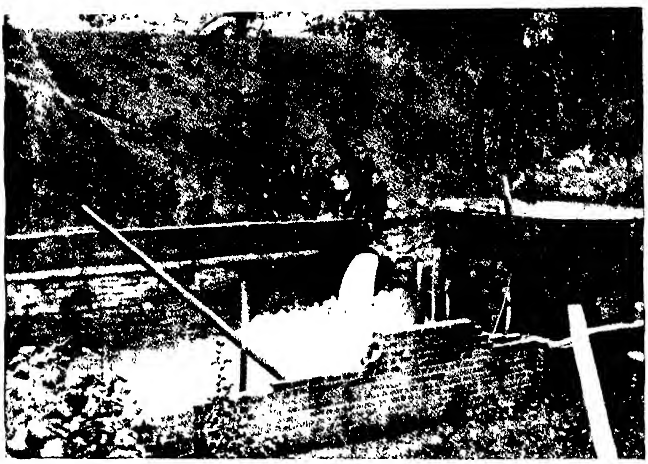


Fig. 85.

pressure on the water surface. It is claimed that this auxiliary pressure prevents surging of the water and thereby steadies the flow from the well. The author has tested the system with and without the outside pressure and finds no appreciable difference in efficiency.

Figure 85 is the flow from a well of the Indianapolis, Ind., Water Works equipped with the Harris system.

Figure 86 is a drawing of an air-lift plant installed by the Harris Air Pump Co., at Shirley, Ind. The well is equipped with the Harris pump and in a pit near the well is installed a Harris

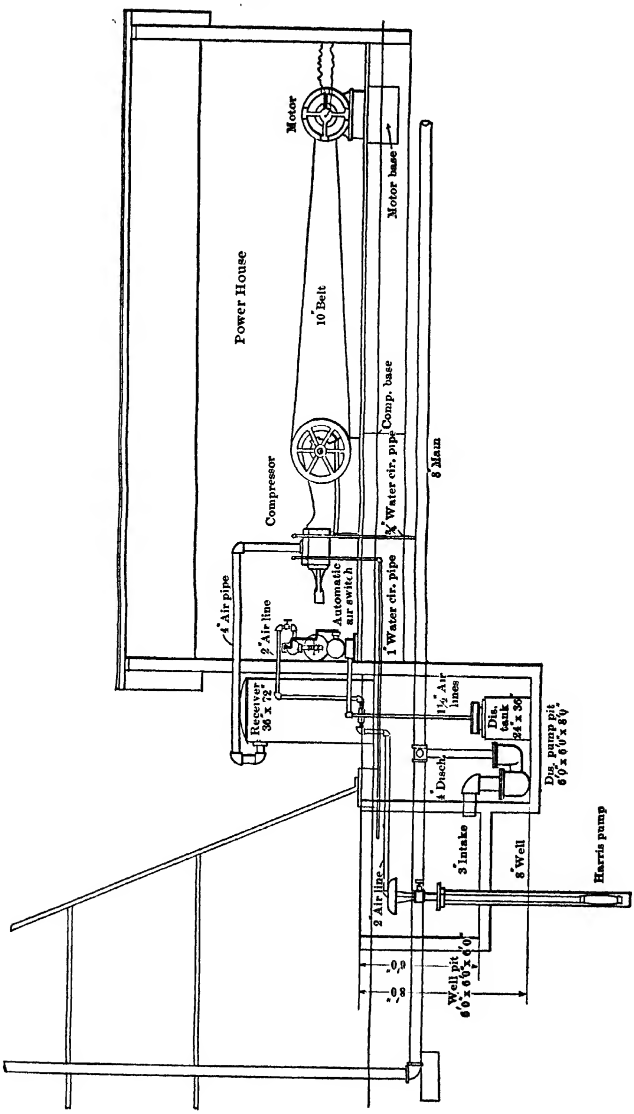

Power House
10" Belt
Motor
Motor base
Comp. base
Compressor
8" Main
¾" Water cir. pipe
1" Water cir. pipe
Automatic air switch
4" Air pipe
2" Air line
Receiver 36" x 72"
1½" Air lines
Dis. tank 24" x 36"
Dis. pump pit 6'0" x 6'0" x 8'0"
4" Disch.
3" Intake
8" Well
Harris pump
2" Air line
Well pit 6'0" x 6'0" x 6'0"
6'0"
8'0"

FIG. 86.

pneumatic displacement pump. The water is pumped from the well and flows by gravity into the displacement pump, and is then pumped up into an overhead tank. The operation is entirely automatic. The motor driving the compressor is fitted with a pressure control, so that variation of water level in the overhead tank starts and stops the system.

For raising water from the well above the surface of the ground the Harris Air Pump Co. sometimes install their booster system. This arrangement consists of an enclosed tank attached to the well top into which the mixture of air and water is pumped. The air is separated from the water and the former, rising to the top of the tank, exerts a pressure on the water surface lifting it the required height. The water volume in the tank is regulated by a float valve. When excessive air accumulates and the water level is lowered, the valve is actuated and the surplus air is allowed to escape to the atmosphere. Fig. 87 shows the arrangement.

The Weber System. — Materially different in construction and operation from any air lift yet described is the system manufactured by the Weber Subterranean Pump Co. The principle of operation is identical with that of the displacement pump. In fact, it is actually a deep-well displacement pump.

Figure 88 is a broken section of the foot piece and Fig. 89 is a diagram of a two-well installation. The return-air principle is here employed and the air is " switched " from well to well and the exhaust from each well is admitted to the compressor suction by the reversing valve shown. When lifts are very high, the system is installed in stages.

Ingersoll-Rand System. — Fig. 90 is an air lift which operates on the principle of the Return Air System previously described. Air and water are discharged vertically into a separator tank which is placed directly over, and attached to, the well. The water flows by gravity to a receiving basin and the air is piped back to the compressor intake.

Any moisture carried in entrainment by the returning air is precipitated in the water leg shown and a valve is provided for bleeding. A check valve mounted on top of the water leg admits whatever make-up air that may be required from time to time.

The advantages claimed for this arrangement are: the salvaging of the remaining air pressure at the point of well discharge which is otherwise dissipated and providing cold air for the

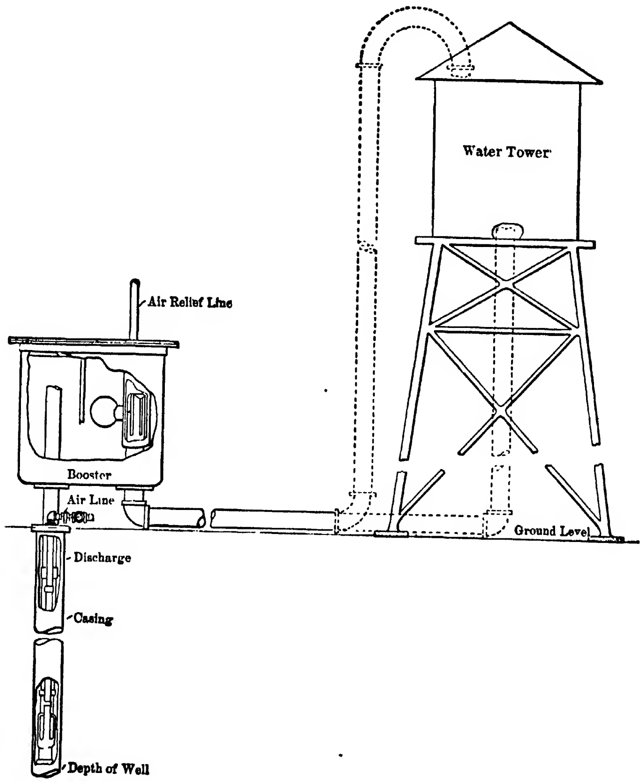


FIG. 87

compressor suction. In other words, the over-all efficiency of the standard air lift is improved by creating advantageous compression conditions for the air cylinder.

The air pressure at the point of discharge of an air lift is equivalent to the velocity head of the column. It is obvious

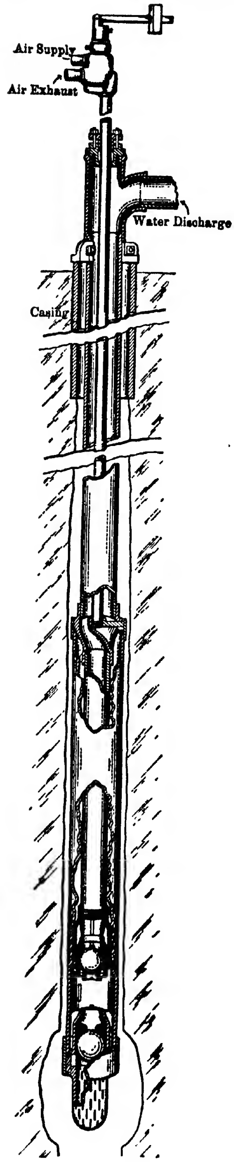


FIG. 88

then, that the superiority of this system becomes more marked as the lift increases.

This general layout was first designed by the writer in 1910 but only recently developed.

Tests. — The author has tested a number of wells equipped with two of the makes of commercial systems described, and the results in some instances show them to be from 10 to 30 per cent more efficient than the systems described in Chapter VII. In other instances the efficiencies were about equal but in no instance was the efficiency lower. Experience seems to indicate that the superiority of the manufactured systems becomes more evident as the lift is increased.

Conclusion. — The advantages obtained by using compressed air as a means of pumping are many, but like all other systems, there are also objectionable features. Summarizing from preceding pages, advantages and limitations are, briefly, as follows:

Advantages — Long Distances. — Owing to the comparatively small losses encountered in transmitting air through properly-designed pipe lines, both the air-lift and the displacement pump may be operated efficiently when located at long distances from the compressor. A larger number of wells or pumps scattered over a considerable area may consequently be operated from a central

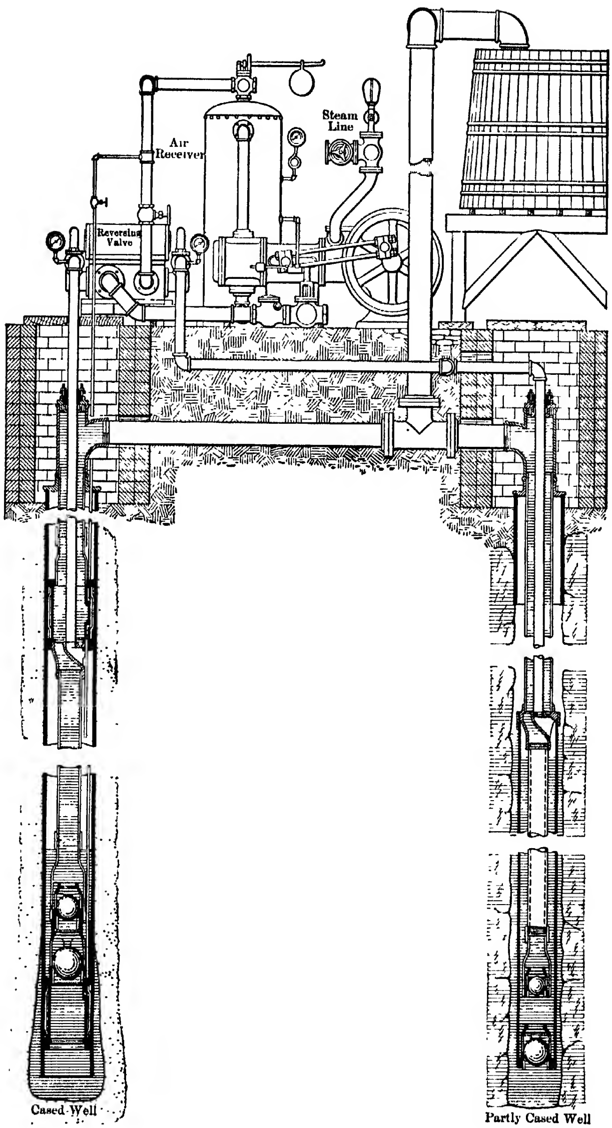

Air
Receiver
Steam
Line
Reversing
Valve
Cased Well
Partly Cased Well

FIG. 89.

plant. This centralization of machinery and effort makes for a considerably lower operating cost than that of independent plants placed at each well which would be necessary if either steam-driven or centrifugal pumps were used.

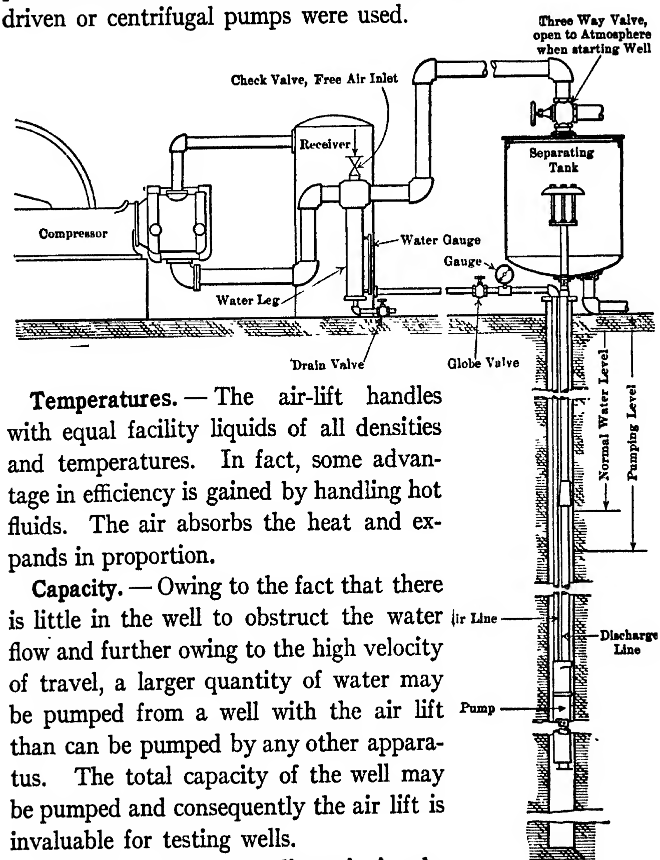


FIG. 90

Temperatures. — The air-lift handles with equal facility liquids of all densities and temperatures. In fact, some advantage in efficiency is gained by handling hot fluids. The air absorbs the heat and expands in proportion.

Capacity. — Owing to the fact that there is little in the well to obstruct the water flow and further owing to the high velocity of travel, a larger quantity of water may be pumped from a well with the air lift than can be pumped by any other apparatus. The total capacity of the well may be pumped and consequently the air lift is invaluable for testing wells.

Aeration. — The expanding air in the discharge pipe abstracts heat from the water and hence lowers the temperature. It has been found that the temperature reduction amounts to from 3 to 5 degrees and sometimes even more. This increases the value of the water for condensing purposes.

Another advantage of thorough aeration is that the quality of water is improved by oxidation of impurities, such as iron. Many instances are on record where well water was unfit for domestic use until after the installation of the air lifts.

Maintenance. — There are no moving parts or wearing surfaces in the well, and, therefore, the cost of upkeep and repairs is negligible. The absence of moving parts makes the air lift particularly adaptable to handling gritty liquids, sewerage, acid or alkaline solutions and, in fact, any liquids or semi-liquids whatever. No form of mechanical deep-well pump can accomplish this without excessive repair cost and expensive shut-downs.

Limitations — Submergence. — One of the most serious handicaps to the air lift is the high percentage of submergence necessary to proper operation. On this account, installation in shallow wells with comparatively high lifts is impracticable. In surface pumping, the difficulty may be overcome by staging the lift, but the small diameter will not permit of staging inside of a well.

Efficiency. — While the actual pumping efficiency of the air lift is admittedly low, still the over-all efficiency figured from the power end of the compressor to water delivered in the reservoir and taking into account upkeep and repairs, compares most favorably with any other means of deep-well pumping.

Surface Pumping. —The losses encountered in transmitting water horizontally and vertically at, and above, the ground surface have already been pointed out. It is unwise to so employ the air lift without making special arrangement as shown and, in fact, the air lift is not adaptable to nor intended for such work.

CHAPTER IX

COMPRESSION GENERALITIES

While it is not the intention to go into the subject of compressed air, and the thermodynamics thereof, elaborately, still there are certain principles and laws that should be stated briefly. On the following pages are given the basic laws and formulæ of air compression that must be recognized when designing and installing compressed-air pumping plants. It is naturally assumed that the reader is familiar with such fundamental definitions and expressions as are necessary to a comprehensive study of the subject.

Boyle's Law.—*At constant temperature the volume of gas is proportional to the absolute pressure, or* $PV = P_1V_1$, where

P = initial absolute pressure in pounds per square inch;
V = initial volume in cubic feet;
P_1 = final absolute pressure in pounds per square inch;
V_1 = final volume in cubic feet.

In other words, the law expresses the fact that if the pressure on a certain enclosed volume of gas is doubled, the volume will be half the original volume (if the temperature is kept constant meanwhile), or conversely, if, at constant temperature, the pressure is reduced by half, the volume will be doubled.

Charles' Law.—*At constant volume the pressure of a perfect gas is directly proportional to the absolute temperature, or at constant pressure the volume is directly proportional to the absolute temperature, or:*

$$\frac{P}{T} = \frac{P_1}{T_1} \text{ and } \frac{V}{T} = \frac{V_1}{T_1}$$

where T and T_1 are initial and final absolute temperatures in degrees F.

Combining Charles' and Boyle's laws, we have the formula

$$\frac{PV}{T} = \frac{P_1V_1}{T_1} \tag{60}$$

Joules' Law. — *When a perfect gas expands, doing no external work, the temperature remains constant.* For instance, in the equation

$$\frac{PV}{T} = \frac{P_1V_1}{T_1}$$

if $T = T_1$, we have $PV = P_1V_1$, which is the law of expansion of a perfect gas.

Specific Heat. — The specific heat of a substance is the amount of heat (B.t.u.) that is required to raise the temperature of one pound of the substance through one degree Fahrenheit.

Specific Heat at Constant Volume C_v. — In the equation

$$\frac{PV}{T} = \frac{P_1V_1}{T_1}$$

if $V = V_1$, then we have $\frac{P}{T} = \frac{P_1}{T_1}$, which is the law of Charles. Suppose we have a certain volume of air contained in a sealed receptacle and the temperature is raised 1° F. The pressure is thereby raised according to the above law, and the intrinsic energy of the air is increased. No work is done, however, because work equals pressure multiplied by distance, and by our supposition, the latter factor is zero. The specific heat at constant volume, then, of air is the amount of heat (B.t.u. or fraction thereof) that is required to raise the temperature of one pound of the air through 1° F., the volume being kept constant as above. C_v for air is found by experiment to be 0.169.

Specific Heat at Constant Pressure C_p. — Assume in this instance, that we have a vertical cylinder containing a quantity of air and resting on the air, is a frictionless piston of constant weight, or pressure P. If the air is heated, the volume will increase, moving the piston outward and external work is performed. The specific heat at constant pressure, then, of air is

the amount of heat (B.t.u. or a fraction thereof) that is required to raise the temperature of one pound of the air through 1° F., if the air is allowed to expand against a constant pressure. Therefore, $C_p = C +$ heat equivalent of external work, and for air has been found to be 0.237. C_p and C_v are measured in B.t.u.'s, so to obtain their equivalents in foot pounds it is necessary to multiply by 778, and the products for convenience of calculation are called K_p and K_v.

Going back to our assumption of the cylinder and piston and assuming further that we have (W) pounds of the air: in order that external work be done and the temperature raised 1° F., it is necessary that W $(C_p - C_v)$ thermal units of heat be applied, or W $(K_p - K_v)$ foot pounds of work. In order to raise the temperature T degrees, W $(K_p - K_v)$ T foot pounds of work must be done on the air. Since work is equal to pressure through volume, we have

$$\text{Work} = PV = W \times (K_p - K_v) \times T°;$$

or assuming $(K_p - K_v) = R$, we have the familiar formula

$$PV = WRT \qquad (61)$$

Theoretically, air may be compressed in two ways — *adiabatically and isothermally.*

Adiabatic Compression *of air is compression without loss of heat.* Consider, for instance, a perfectly insulated cylinder and piston having a full charge of air between piston and cylinder head. As the piston advances, the volume of air becomes smaller and the temperature rises, the former in inverse proportion to the absolute pressure exerted and the latter equivalent to the amount of work done. Under these conditions the air at the end of compression will retain all the heat so produced, and this particular compression is called adiabatic. In actual practice such conditions of compression are impossible.

In adiabatic compression the law $\frac{P}{P_1} = \frac{V_1}{V}$ is not followed strictly because as the temperature rises unchecked, it reacts on

the air being compressed to increase the volume. Therefore, to write an expression for adiabatic compression, it is necessary that $\frac{V_1}{V}$ be increased by an amount equivalent to the amount of external work done on the air by heat reaction during compression. It has been shown in various works on thermodynamics that

$\frac{P}{P_1} = \frac{(V_1)^n}{(V)^n}$ where $n = \frac{C_p}{C_v}$ = 1.406 for air holds nearly true. (See Perry's work on the Steam engine.)

Work of Adiabatic Compression. — Fig. 91 shows a theoretical indicator card of an air cylinder having no clearance. The total work done is equal to the work of compression shown by the area under the curve *bc*; *plus* the work of expulsion of the air from the cylinder shown by the area P_2V_2; *minus* the work done on the piston by the intake air shown by the area P_1V_1. Then, calling Q the total amount of work,

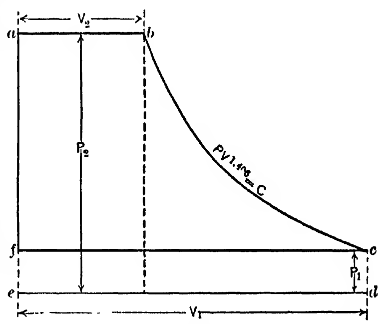


FIG. 91.

$$Q = 498.7\, P_1V_1\left(\left[\frac{P_2}{P_1}\right]^{.29} - 1\right) \tag{62}$$

and the horse power required to compress one cubic foot of free air per minute adiabatically is

$$\text{H.P.} = \frac{1}{4.5}\left(\left[\frac{P_2}{P_1}\right]^{.29} - 1\right) \tag{63}$$

Isothermal Compression. — *Isothermal compression is compression at constant temperature.* In other words, it is compression wherein all heat is removed by some form of cooling device as fast as it is produced. The relation, then, existing

between pressure and volume at any instant is shown by the equation

$$P_1V_1 = P_2V_2 = C \tag{64}$$

Work of Isothermal Compression. — Fig. 92 is the theoretical indicator diagram of isothermal compression in a cylinder having no clearance. Compression begins as before at absolute pressure P_1 and volume V_1 and ends at P_2 and V_2. The total work, Q, in foot pounds done on the air is equal to the algebraic sum of the work of compression, expulsion, and the work done by the intake air, and is shown in the equation

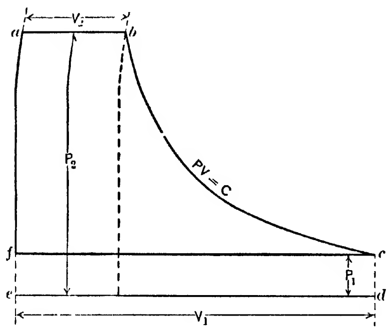


FIG. 92.

$$Q = 144\, P_1V_1 \log_e\left(\frac{P_2}{P_1}\right) \tag{65}$$

and the horse power required to compress one cubic foot of free air per minute isothermally is

$$\text{H.P.} = \frac{1}{15.6} \log_e\left(\frac{P_2}{P_1}\right) \tag{66}$$

Actual Compression with Clearance. — In the every-day practice of air compression, neither of the two formulæ would apply, for it is impossible to design a cylinder wherein either adiabatic or isothermal compression can be obtained. The cylinder in which we are interested is equipped with a water jacket for the removal of some of the heat of compression and to facilitate lubrication, but *all* the heat cannot be so removed. A certain amount is retained by the air itself, and some is left in the piston and cylinder walls. The actual compression curve, then, will lie somewhere between the isothermal and the adiabatic curves, and the exact location depends upon the efficiency of the water jacket, the temperature of the circulating water, etc.

Also in the actual cylinder there is a certain amount of lost space, or clearance, in and around the inlet and discharge valves, and between the piston and cylinder heads at the end of the stroke, all of which has its effect upon the shape of the indicator card. This space at the end of the stroke of the piston is filled with air at the discharge pressure and temperature. As the piston recedes the air expands, doing work on the piston, and finally, the reëxpanded air occupies part of the volume of the cylinder behind the piston. No air can be drawn into the cylinder until the pressure inside falls below that of the atmosphere.

Because of the irregularities that occur in each instance, it is quite impossible to deduce a formula that will cover all conditions of practice. We can, however, derive results that are quite interesting from a theoretical viewpoint, and which will be a guide in the design of the compressor.

In the compression of a perfect gas receiving heat in some regular way, the following relation of volume and pressure at any instant holds nearly true

$$PV^n = C \tag{67}$$

The value of n on the ordinary single-stage air compressor is given by Church as 1.33 and by Unwin as 1.25. The true value, however, will vary in each instance and is dependent upon the size of cylinder, speed of machine, design of water jacket and temperature and amount of cooling water.

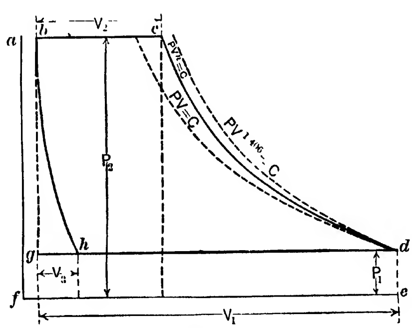


FIG. 93.

In Fig. 93 is shown an air diagram taken from a cylinder with clearance, whose compression curve lies between the isothermal and adiabatic curves. The total amount of work done during the forward stroke is shown by the area $bcdg$ (q), but by

the reëxpansion of the clearance air, there is an amount of work q_1 returned to the receding piston, shown by the area *bhg*. Therefore, the net amount done by the piston is shown by the area *bcdh*, or Q, and its value is given in the expression,

$$Q = \frac{P_1 (V_1 - V_3) n}{n - 1} \left(\left[\frac{P_2}{P_1} \right]^{\frac{n-1}{n}} - 1 \right) \qquad (68)$$

Now $(V_1 - V_3)$ is the net amount of air drawn into the cylinder. The horse power required to compress one cubic foot of free air per minute is shown by

$$\text{H.P.} = \frac{n}{15.6 (n - 1)} \left(\left[\frac{P_2}{14.7} \right]^{\frac{n-1}{n}} - 1 \right) \qquad (69)$$

Figure 94 shows the actual air indicator card taken from a single-stage, straight-line compressor, having poppet inlet and discharge valves. The areas A and B represent the amount of work necessary to open the discharge and the inlet valves, and *ab* is the volume occupied by the reëxpanded clearance air. The volume lying between the suction line and the atmospheric line is the energy expanded to fill the cylinder with air.

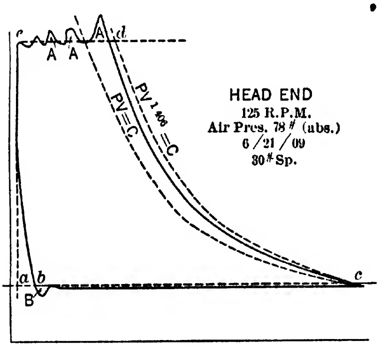


FIG. 94.

Tables 16 and 17 show the horse power, etc., required to compress air from 14.7 pounds per square inch in pressure and 60° F. up to various pressures, in both single- and two-stage compressors, employing both adiabatic and isothermal compression.

Two-stage Compression. — It is evident now, that isothermal compression requires the expenditure of the least amount of work. As before shown, this form of compression is impossible

TABLE 16

Horse Power Required for Compressing One Cubic Foot of Free Air per Minute (Isothermally and Adiabatically) from Atmospheric Pressure (14.7 Pounds) to Various Gauge Pressures

Single-stage Compression

Initial Temperature of Air Taken as 60° F. — Jacket Cooling Not Considered

Gauge pressure, pounds	Absolute pressure, pounds	Number of atmospheres	Isothermal compression		Adiabatic compression			
			Mean effective pressure	H.P.	Mean effective pressure, theoretical	Mean effective pressure plus 15 per cent friction	H.P., theoretical	H.P. plus 15 per cent friction
5	19.7	1.34	4.13	0.018	4.46	5.12	0.019	0.022
10	24.7	1.68	7.57	0.033	8.21	9.44	0.036	0.041
15	29.7	2.02	11.02	0.048	11.46	13.17	0.050	0.057
20	34.7	2.36	12.62	0.055	14.30	16.44	0.062	0.071
25	39.7	2.70	14.68	0.064	16.94	19.47	0.074	0.085
30	44.7	3.04	16.30	0.071	19.32	22.21	0.084	0.096
35	49.7	3.38	17.90	0.078	21.50	24.72	0.094	0.108
40	54.7	3.72	19.28	0.084	23.53	27.05	0.103	0.118
45	59.7	4.06	20.65	0.090	25.40	29.21	0.111	0.127
50	64.7	4.40	21.80	0.095	27.23	31.31	0.119	0.136
55	69.7	4.74	22.95	0.100	28.90	33.23	0.126	0.145
60	74.7	5.08	23.90	0.104	30.53	35.10	0.133	0.153
65	79.7	5.42	24.80	0.108	32.10	36.91	0.140	0.161
70	84.7	5.76	25.70	0.112	33.57	38.59	0.146	0.168
75	89.7	6.10	26.62	0.116	35.00	40.25	0.153	0.175
80	94.7	6.44	27.52	0.120	36.36	41.80	0.159	0.182
85	99.7	6.78	28.21	0.123	37.63	43.27	0.164	0.189
90	104.7	7.12	28.93	0.126	38.89	44.71	0.169	0.195
95	109.7	7.46	29.60	0.129	40.11	46.12	0.175	0.201
100	114.7	7.80	30.30	0.132	41.28	47.46	0.180	0.207
110	124.7	8.48	31.42	0.137	43.56	50.09	0.190	0.218
120	134.7	9.16	32.60	0.142	45.69	52.53	0.199	0.229
130	144.7	9.84	33.75	0.147	47.72	54.87	0.208	0.239
140	154.7	10.52	34.67	0.151	49.64	57.08	0.216	0.249
150	164.7	11.20	35.59	0.155	51.47	59.18	0.224	0.258
160	174.7	11.88	36.30	0.158	53.70	61.80	0.234	0.269
170	184.7	12.56	37.20	0.162	55.60	64.00	0.242	0.278
180	194.7	13.24	38.10	0.166	57.20	65.80	0.249	0.286
190	204.7	13.92	38.80	0.169	58.80	67.70	0.256	0.294
200	214.7	14.60	39.50	0.172	60.40	69.50	0.263	0.303

TABLE 17

Horse Power Required for Compressing One Cubic Foot of Free Air per Minute (Isothermally and Adiabatically) from Atmospheric Pressure (14.7 Pounds) to Various Gauge Pressures

Two-stage Compression

Initial Temperature of Air Taken as 60° F. — Jacket Cooling Not Considered

Gauge pressure, pounds	Absolute pressure, pounds	Number of atmospheres	Correct ratio of cylinder volume	Intercooler gauge pressure	Isothermal compression		Adiabatic compression				Percentage of saving over one-stage compression
					Mean effective pressure	H.P.	Mean effective pressure, theoretical	Mean effective pressure plus 15 per cent friction	H.P. theoretical	H.P. plus 15 per cent friction	
50	64 7	4 40	2 10	16 2	21 80	0 095	24.30	27 90	0 106	0 123	10 9
60	74 7	5 08	2 25	18 4	23 90	0 104	27 20	31 30	0.118	0.136	11 3
70	84 7	5 76	2 40	20 6	25 70	0 112	29 31	33 71	0 128	0 147	12 3
80	94 7	6 44	2 54	22 7	27 52	0 120	31 44	36 15	0 137	0 158	13 8
90	104 7	7 12	2 67	24 5	28 93	0 126	33 37	38.36	0 145	0.167	14.2
100	114 7	7 80	2 79	26 3	30 30	0 132	35.20	40 48	0 153	0 176	15 0
110	124 7	8 48	2 91	28 1	31 42	0 137	36 82	42 34	0 161	0 185	15 2
120	134 7	9 16	3 03	29 8	32 60	0 142	38 44	44 20	0 168	0 193	15 6
130	144 7	9 84	3 14	31 5	33 75	0 147	39 86	45.83	0 174	0 200	16 3
140	154 7	10 52	3 24	32 9	34 67	0 151	41.28	47 47	0 180	0 207	16 7
150	164 7	11 20	3 35	34 5	35 59	0.155	42 60	48 99	0.186	0.214	16 9
160	174 7	11 88	3 45	36 1	36 30	0 158	43 82	50.39	0 191	0 219	18 4
170	184 7	12 56	3 54	37 3	37 20	0 162	44 93	51 66	0 196	0 225	19 0
180	194 7	13 24	3 64	38 8	38 10	0 166	46 05	52.95	0 201	0.231	19 3
190	204 7	13 92	3 73	40 1	38 80	0 169	47.16	54 22	0 206	0 236	19 5
200	214 7	14 60	3 82	41.4	39 50	0 172	48 18	55 39	0.210	0 241	20 1
210	224 7	15 28	3 91	42 8	40 10	0 174	49 35	56 70	0 216	0 247	. . .
220	234 7	15 96	3 99	44 0	40 70	0 177	50 30	57 70	0.220	0.252	. . .
230	244 7	16 64	4 08	45 3	41 30	0 180	51 30	59 10	0.224	0 257	......
240	254 7	17 32	4 17	46 6	41 90	0 183	52 25	60 10	0 228	0 262	. .
250	264.7	18 00	4 24	47 6	42 70	0 186	52 84	60.76	0.230	0.264	.
260	274 7	18 68	4 32	48 8	43 00	0 188	53.85	62.05	0.235	0 270	.. . .
270	284 7	19 36	4 40	50.0	43 50	0 190	54.60	62.90	0.238	0 274	.....
280	294 7	20.04	4 48	51.1	44.00	0 192	55 50	63 85	0 242	0 278	.
290	304.7	20 72	4 55	52 2	44 50	0 194	56.20	64 75	0.246	0 282	......
300	314 7	21.40	4 63	53.4	45 30	0 197	56.70	65 20	0 247	0.283	.... .
350	364 7	24 80	4 98	58 5	47.30	0 206	60.15	69.16	0.262	0 301	. ...
400	414 7	28.20	5.31	63 3	49 20	0 214	63.19	72.65	0.276	0 317	......
450	464 7	31 60	5.61	67.8	51.20	0 223	65.93	75 81	0 287	0.329	.. . .
500	514 7	35 01	5.91	72 1	52 70	0 229	68 46	78 72	0 298	0 342	..

in practice, but a material saving can be realized by compressing in stages and cooling the air between each stage. In this way isothermal compression is approached, as will be more fully seen later.

In two-stage compression the air is drawn from the atmosphere into the first, or low-pressure, cylinder, and there compressed up to a certain point. It is then forced through an intercooler where the temperature is reduced by circulating water and thence drawn into the second or high-pressure cylinder where compression is continued up to the desired terminal pressure.

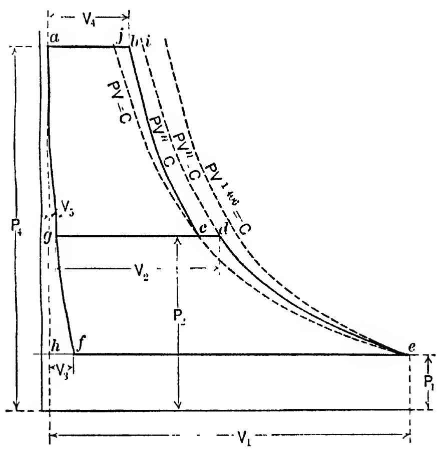


FIG. 95.

Work of Two-stage Compression. — An attempt is made in the design of two-stage compressors to divide the work equally between the two cylinders. Very often, however, actual working conditions are different from those contemplated, consequently, the equality is destroyed. In the following discussion it is assumed that the work is the same in each cylinder, and further, that the temperature of the air after passing through the intercooler is the same as that of the atmosphere.

Figure 95 shows the cycle of operations of a two-stage machine designed as above. A volume V_1 of air under pressure P_1 is

drawn into the low-pressure cylinder and there compressed to volume V_2 and pressure P_2. The air is cooled and the volume is reduced to that shown by *gc*, which is equivalent to the volume obtained in isothermal compression from P_1 to P_2. The high-pressure cylinder then receives the air and compresses it up to pressure P_4 and volume V_4. The curve of compression follows the broken line *edcb*.

Now, if the air were compressed in a single-stage machine from P_1 to P_4 the curve of compression would be *ei* ($PV^n = C$), and the work done evidenced by the area *aief*. The work done in two-stage compression is shown by the area *abcdef* and the saving realized over single-stage compression is shown by the area *bidc*.

Let Q_1 and Q_2 = work in foot pounds to compress air in the low- and high-pressure cylinders, respectively; and Q = total work of compression. The value of Q, then is

$$Q = 288 \frac{P_1(V_1 - V_3)n}{n - 1}\left(\left[\frac{P_2}{P_1}\right]^{\frac{n-1}{n}} - 1\right) \tag{70}$$

The horse power required to compress one cubic foot of free air per minute in this way, remembering that $(V_1 - V_3)$ is the net amount of air drawn into the low-pressure cylinder is

$$\text{H.P.} = \frac{n}{7.8\,(n - 1)}\left(\left[\frac{P_2}{14.7}\right]^{\frac{n-1}{n}} - 1\right)^{*} \tag{71}$$

Figure 96 shows a combined air card taken from a $8\frac{1}{2}$-inch and $12\frac{1}{2}$-inch and 14 by 16-inch two-stage air-end, duplex-steam end compressor operating at 115 r.p.m., and against a pressure of 195 pounds (abs.). At the time the cards were taken, the compressor was furnishing air for oil well No. 12, owned by the Crowley Oil & Mineral Co., at Evangeline, La.†

* It must be remembered that P_2 in Eq. 71 is the intercooler pressure while in formulæ for work done, etc., in single-stage compression P_2 designates terminal pressure.

† See Transactions of A.S.M.E., Vol. 31, *Tests Upon Compressed Air Pumping Systems of Oil Wells*, by E. M. Ivens.

The shaded area shown represents work lost, for, obviously, it has to be performed twice. The reduction in pressure of the air at the point of valve opening in the high-pressure cylinder is caused by cooling in the intercooler, and frictional losses through

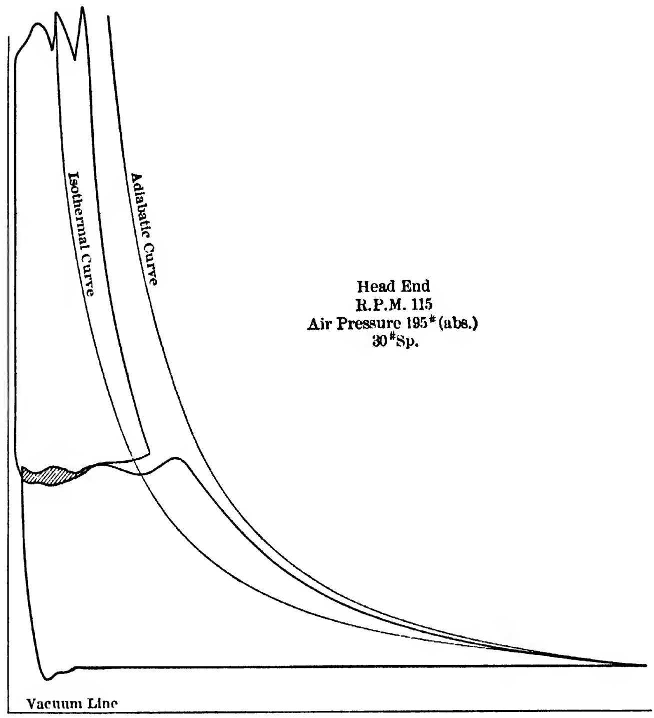


FIG. 96.

the valves and intercooler. The almost steady rise in pressure during the suction stroke of the high-pressure cylinder is probably caused by heating of the air by the already hot valves, cylinder walls, piston, etc.

Lengthy computations and derivations of formulæ for work done in air compression are of little value in actual practice, but considerable benefit is derived from the study of theoretical

conditions in any branch of engineering. Our formulæ cannot be applied with any degree of satisfaction to the solution of practical problems because we have employed factors whose values vary, not only in each case, but for nearly every instant in any one case. The value of n, for instance, is dependent upon a number of varying things, as before stated, and, consequently, the results obtained by the use of formulæ are unsatisfactory.

In our derivations, we have assumed that the intake air was at a temperature of 60° F. and a pressure of 14.7 pounds per square inch. These will vary with the season of the year, the location and the altitude, but volumes of intake air at varying temperatures and pressure observed may be readily converted to equivalent volumes at 60° F. and 14.7 pounds pressure by the application of the foregoing laws.

Table 18, taken from Hiscox's *Compressed Air*, shows the loss in capacity, etc., of compressors operating at various altitudes and will be found quite useful in air compressor capacity calculations.

TABLE 18

EFFICIENCY OF COMPRESSORS AT DIFFERENT ALTITUDES

(From Hiscox's *Compressed Air*)

Altitude in feet	Barometric pressure		Volumetric efficiency of compressor, per cent	Loss of capacity, per cent	Decreased power, per cent
	Inches of mercury	Pounds per square inch			
0	30 00	14 75	100	0	0 0
1,000	28 88	14.20	97	3	1.8
2,000	27 80	13.67	93	7	3.5
3,000	26 76	13 16	90	10	5.2
4,000	25 76	12.67	87	13	6.9
5,000	24 79	12.20	84	16	8.5
6,000	23 86	11 73	81	19	10.1
7,000	22.97	11 30	78	22	11.6
8,000	22 11	10 87	76	24	13.1
9,000	21.29	10.46	73	27	14 6
10,000	20 49	10 07	70	30	16.1
11,000	19.72	9 70	68	32	17.6
12,000	18.98	9.34	65	35	19.1
13,000	18.27	8.98	63	37	20.6
14,000	17.59	8.65	60	40	22.1
15,000	16 93	8.32	58	42	23.5

Single- versus Two-stage Compression. — It must not be assumed that two-stage compressors are always more preferable than single-stage ones. Oft-times pressure requirements are such that the latter are more economical in operation as well as more attractive in first cost and floor-space occupancy. Multi-stage compressors under all conditions have a higher compression efficiency than single-stage machines of like capacities, but the ultimate or over-all efficiency of the latter is greater for terminal pressures of 80 pounds (gauge) and under, and it is this latter efficiency that the operator is most interested.

A two-stage compressor requires, in addition to the parts of the single-stage machine, a high-pressure cylinder, valves and piston; an intercooler piped to the cylinders; and sometimes pumps for circulating water through the intercooler. This additional apparatus complicates the machine, increases the cost and floor space required, and increases the mechanical friction over the single-stage compressor. When the interest on the additional first cost *plus* the extra maintenance cost *plus* the mechanical efficiency loss is greater than the gain due to cooling during compression, then naturally it is unwise to install a two-stage machine. This condition does exist for all pressure requirements of 80 pounds and under, as before said.

Shortly after the air leaves the compressor cylinder, it is cooled to the temperature of the atmosphere. This means that the heat compression has been dissipated, and the energy or work necessary to its creation lost. The object of multi-stage compression is to minimize this loss, which will ultimately occur, by removing heat of compression during compression itself, and thereby reducing the total amount of work required of the machine; all of which means a smaller expenditure of energy and, consequently, cheaper operation. If it were possible to use the air without first losing the heat of compression any cooling during compression would, obviously, be a loss to efficiency.

The ideal compressor, then, is one in which all heat of compression is removed as fast as it is generated. To realize this in practice, it would be necessary that the machine have an infinite

number of stages, and that the air be cooled to its initial temperature between each stage. Fig. 97 is a diagram showing the horse power required to compress 100 cubic feet of free air per minute

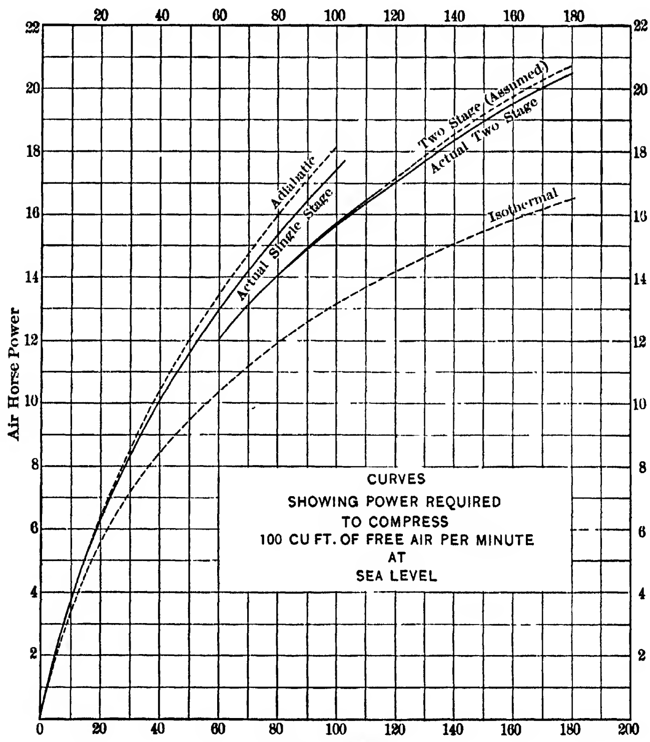


FIG. 97.

up to various gauge pressures. The "Actual Single-stage" and the "Actual Two-stage" compression curves were plotted with average results obtained from a large number of tests of various makes and designs of air compressors. The "Two-stage (Assumed)" curve was plotted from calculated results, assuming

frictionless transmission of air between the cylinders and adiabatic compression in each cylinder, together with other assumptions previously referred to in the derivation of work formulæ.

Air Compression at Altitudes.—In our discussion thus far, we have assumed that the compressor is being operated at sea level; that is that the inlet air is under an absolute pressure of 14.7 pounds per square inch. If the machine is operated at a greater altitude, the intake air pressure will be proportionately less and additional work is imposed upon the compressor. This fact is self-evident.

The capacity of a given compressor is less at higher altitudes than at sea level because of the diminished density of the intake air. In other words, at every stroke of the machine a smaller mass, or weight, of air is drawn into the cylinder. This should be kept well in mind and due allowance made when choosing a compressor to perform a certain duty.

Volumetric efficiency is also less at altitudes due to the fact that the clearance air expands to the lower atmospheric pressure, and, consequently, when expanded occupies a larger volume of the cylinder.

Stage compression can be employed to greater advantage at high altitudes because the heat of compression, which increases with the ratio of the final to the initial absolute pressures, is greater. Stage compression at altitudes is justifiable for pressure considerably under 80 pounds.

Table 19 gives the multipliers for determining the volume of free air at various altitudes which, when compressed to various pressures, is equivalent in effect to a given volume of air at sea level.

TABLE 19

MULTIPLIERS FOR DETERMINING THE VOLUME OF FREE AIR AT VARIOUS ALTITUDES WHICH, WHEN COMPRESSED TO VARIOUS PRESSURES, IS EQUIVALENT IN EFFECT TO A GIVEN VOLUME OF FREE AIR AT SEA LEVEL

Altitude in feet	Barometric pressure		Multiplier				
			Gauge pressure (pounds)				
	Inches of mercury	Pounds per square inch	60	80	100	125	150
0	30 00	14.75	1 000	1 000	1.000	1.000	1 000
1,000	28 88	14 20	1.032	1 033	1.034	1.035	1.036
2,000	27.80	13.67	1.064	1.066	1.068	1.071	1.072
3,000	26.76	13.16	1.097	1.102	1.105	1.107	1.109
4,000	25.76	12.67	1.132	1.139	1.142	1.147	1.149
5,000	24.79	12 20	1.168	1.178	1.182	1.187	1.190
6,000	23.86	11.73	1.206	1.218	1.224	1.231	1.234
7,000	22 97	11.30	1.245	1.258	1.267	1.274	1.278
8,000	22.11	10.87	1.287	1.300	1.310	1.319	1.326
9,000	21.29	10.46	1.329	1.346	1.356	1.366	1.374
10,000	20.49	10.07	1.373	1.394	1.404	1.416	1.424

CHAPTER X

THE AIR CARD AND AIR COMPRESSOR EFFICIENCY

The indicator is indispensable in steam-engine design and operation, but it is, if possible, even more valuable in air-compressor practice. In the former the field of usefulness is somewhat limited in-so-far as the determining of the mechanical efficiency of the machine is concerned. That is, if the mechanical efficiency of an engine is to be determined, it is necessary to employ some form of absorption dynamometer in connection with the indicator; then the brake horse power, as observed from the former divided by the indicated horse power, is the mechanical efficiency, and the difference between the two is the power necessary to overcome the friction of the machine itself. Sometimes friction diagrams are taken, *i.e.*, cards taken when there is no load on the engine other than its own friction, and their area subtracted from the full-load card, and efficiency computed. This method is inaccurate because the friction is obviously much greater throughout all bearings when the engine is loaded than when merely turning over with no external load.

With the air compressor these limitations do not exist, as cards taken simultaneously from steam and air cylinders are full statements of the power conversion. The steam cards show the amount of energy put into the machine and the air cards show the power delivered in return, and the difference between the two is an accurate statement of the friction. By taking cards from the compressor when operating under varying loads, the friction for every change of load may be accurately determined. Cards taken simultaneously also show the relation between power and resistance at every point in the stroke.

It will be a little difficult at first for those who are accustomed to reading steam engine diagrams to examine intelligently

cards from an air cylinder. One can best learn by keeping in mind that the one is the direct opposite of the other. In other words, the steam diagram is the record of pressures of an expanding gas doing external work, while the air diagram is the record of pressures of a gas being compressed and having work done

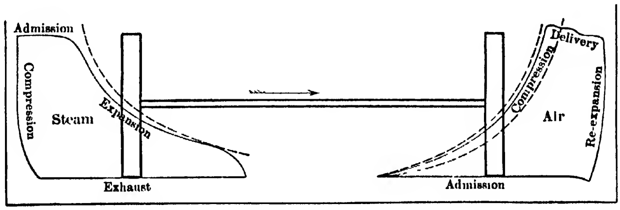


FIG. 98.

on it. Therefore, the expansion line of the steam diagram corresponds to the compression line of the air diagram; the admission line of the one corresponds to the discharge line of the other; and the compression line corresponds to the reëxpansion line, and all as clearly shown in Fig. 98.

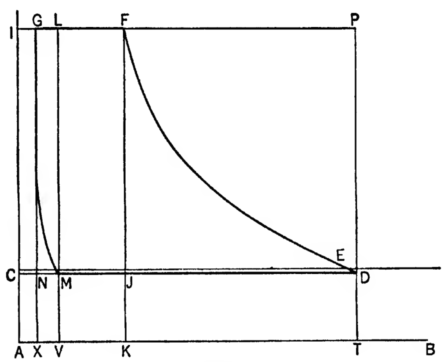


FIG. 99.

Figure 99 is a diagram from the "air end" of a single-stage air compressor. The lines are smoother and more nearly perfect than those of the actual card. The atmospheric, the vacuum and

the clearance lines are all located and drawn in exactly the same manner as for the steam diagram. In Fig. 99 the atmospheric line is purposely drawn low in order to distinguish it from the admission line. In actual diagrams, the admission line does fall below the atmospheric line varying distances. In well-designed cylinders this pressure difference averages $\frac{1}{4}$ pounds, while in poorly-designed cylinders it is as great as 1.5 pounds. Restricted port areas, long intake pipes and heavy inlet valve spring increase this loss.

A volume of air, represented by the rectangle *MLPD*, is drawn into the cylinder and compressed from the absolute pressure, represented by *TD*, up to that represented by *FK*. In so doing, the volume has been reduced to *GFJN*, and this is delivered to the receiver or pipe lines. The actual piston displacement is represented by the rectangle *NGPD*. There is, necessarily, in every cylinder a certain amount of clearance between the piston and the head and around the valves. These clearance spaces, represented on the diagram by rectangle *NCIG*, at the end of the stroke of the piston, are filled with air at the discharge pressure. As the piston recedes, this clearance air expands along the line *GM* (which is practically adiabatic), until it finally occupies the volume *MCIL*, when admission of air from the atmosphere begins and the cycle is repeated.

Air compressors are rated by all manufacturers according to their piston displacement, and, consequently, to ascertain the actual capacity of a compressor one must know the volumetric efficiency of its air cylinder. We shall discuss this at greater length later.

Theoretical Curves. — To form a comparison of actual compression of air and compression under ideal conditions, it is necessary to draw the theoretical curves. This may be done in exactly the same manner as for steam, and, usually, both the adiabatic and the isothermal curves are drawn on the air diagram. To facilitate the drawing of these curves, Mr. Frank Richards, on pages 48 and 49 of his *Compressed Air*, has provided diagrams which are very useful. Mr. H. V. Conrad, in *Power*,

March, 1911, has compiled very convenient and accurate tables for laying off theoretical air-compression curves.

While it is well to appreciate the value of the air diagram, still it must not be trusted blindly, for it often conveys false impressions, and, in skillful hands, the indicator can be made to tell some very flattering things. For instance, if longitudinal by-passes were cut in the cylinder walls near the heads and slightly longer than the thickness of the piston, so that at the end of the stroke

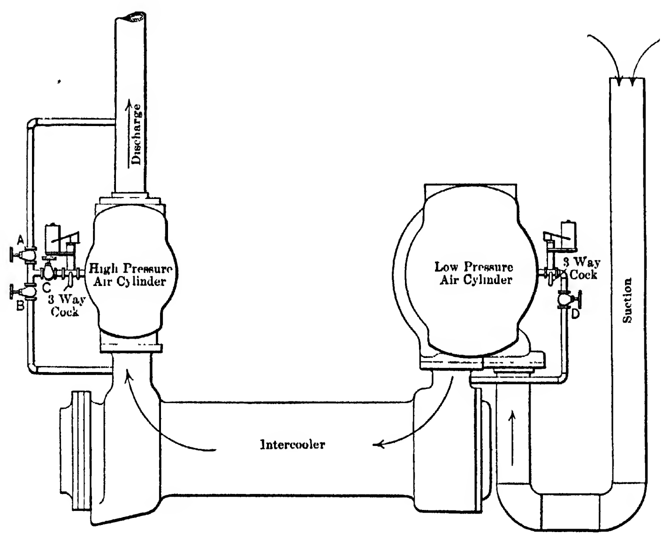


FIG. 100.

the piston uncovers their ends, the clearance air under the discharge pressure will escape to the opposite side of the piston, and a card taken will show a very high volumetric efficiency. A scored cylinder will give a card whose compression curve is much nearer the isothermal than ordinary, as will a leak by the suction valves. The delivery line of the air card is wavy and irregular, due to the action of the valves, so it is difficult to determine with accuracy the discharge pressure. In Fig. 100 are shown indicators

attached to the air cylinders of a two-stage compressor. The pipe connections are such that, by manipulating the valves, discharge pressure lines may be drawn on all the diagrams and both discharge and inlet pressure lines drawn on the high-pressure cylinder diagram.

Air-compressor Efficiencies *— Mechanical Efficiency. — The energy in the steam admitted to the steam cylinder of an air compressor is expended in the following ways:

1. To heat the steam-cylinder walls and piston;
2. To compress the air;
3. To heat the air during compression;
4. To heat the jacket water;
5. To overcome the friction of the machine.

The horse power required by 2 and 3 may be computed from the air-indicator diagram; 4 may be found by observing the temperature of entering and leaving jacket water, together with its weight and computing the B.t.u.'s therefrom, and the equivalent horse power; and 5 is found by subtracting the sum of the first three from the indicated horse power in the steam cylinder.

The Mechanical Efficiency of a steam-driven air compressor, then, is equal to the air horse power *plus* the jacket horse power divided by the indicated horse power, or

$$E_m = \frac{\text{A.H.P.} + \text{Jkt. H.P.}}{\text{I.H.P.}} \tag{72}$$

and the mechanical efficiency of a power-driven machine is expressed by

$$E_m = \frac{\text{A.H.P.} + \text{Jkt. H.P.}}{\text{Brake H.P. delivered to compressor shaft}} \cdot \tag{73}$$

This efficiency depends upon the mechanical construction of the machine and the lubrication. It will be found to vary from 75 per cent in poorly-designed machines up to 92 per cent in the best designs.

* *Air Compressor Efficiencies*, by E. M. Ivens in *Power*, Oct. 15, 1912.

Compression Efficiency. — Compression (or compressor) efficiency is the ratio of the theoretical horse power required to compress an amount of air to that actually required, or

$$E_c = \frac{\text{theoretical H.P.}}{\text{A.H.P.}} \tag{74}$$

The adiabatic and the isothermal horse powers are both theoretical, but since the term efficiency is a statement of how nearly perfect a machine or device is, it is proper that we use the latter (referring to no clearance base) in our formula.

This efficiency depends upon the design of water jacket and cooling appliances, and it is principally to increase compression efficiency that multi-stage compression is employed.

To determine the compression efficiency, the isothermal curve is plotted on the air card (Fig. 101), starting, of course, at the beginning of the stroke, and ending at the theoretical delivery line, or terminal pressure line. The area *abdf* thus enclosed divided by the area *acde* of the actual card is the compression efficiency. Indicator cards that show a very high compression efficiency should be looked upon with suspicion, as investigation will invariably show that either the suction valves leak or air is escaping from the compression side of the moving piston to the suction side, due to scored cylinder or leaky piston. Actual compression curves will follow the adiabatic curve quite closely as the water jacket has little effect other than to facilitate lubrication.

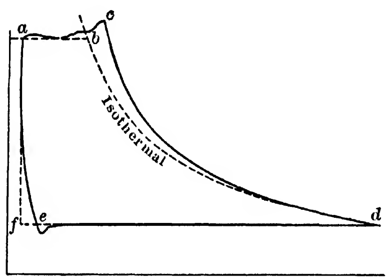


FIG. 101.

Volumetric Efficiency. — Volumetric efficiency is the ratio of the actual number of cubic feet of free air compressed per unit of

time, to the number of cubic feet of piston displacement diving that time, or

$$E_v = \frac{\text{Actual cubic feet of free air per minute}}{\text{Cubic feet of piston displacement per minute}} \qquad (75)$$

On the indicator diagram the observed volumetric efficiency is (Fig. 102) obviously $\frac{ad}{bc}$.

Volumetric Efficiency Depends, First, upon the Clearance Volume in the Air Cylinder. — If there were no clearance between cylinder heads and piston at the end of the stroke, and no lost space in and around the valves, the volumetric efficiency (referring to atmospheric air) would always be 100 per cent. The greater the clearance volume, then, the greater will be the volume of the cylinder occupied by the expanded clearance air. This fact is self-evident.

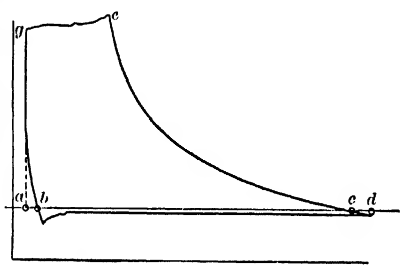


FIG. 102.

Volumetric Efficiency Depends, Second, upon the Terminal Pressures. — The higher the terminal pressure of air in any given cylinder, the greater will be the volume occupied by the expanded air of the clearance spaces. This means that, as the terminal pressure is increased, the volumetric efficiency decreases. To show this graphically, there are reproduced in Fig. 103 three super-imposed diagrams taken from the same cylinder at the different pressures shown. The increase in volumetric efficiency as the terminal decreases is plainly evident in the illustration.

Volumetric Efficiency Depends, Third, upon the Temperature and Pressure of the Intake Air. — Since, by our definition, volumetric efficiency refers to free air, or air at 14.7 pounds pressure, and 60° F., then every change of temperature and

pressure of initial (intake) air has its effect directly upon the volumetric efficiency. For instance, let us suppose we have a room whose temperature is 60° F., and whose atmospheric pressure is 14.7 pounds or, in other words, conditions where actual *Free Air* is available and drawn without heating into the cylinder. In this room is located a compressor whose capacity is 135 cubic feet of this air per minute, whose piston displacement is 150 cubic feet per minute and whose terminal pressure is 100 pounds. The volumetric efficiency under these conditions, then, is

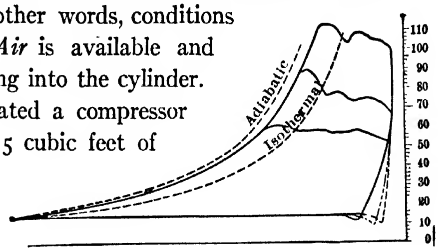


FIG. 103.

$$E_v = \frac{135}{150} = 90 \text{ per cent} \tag{76}$$

Now, suppose that from some cause the temperature of intake air were raised from 60° F. to, say, 65° F., the atmospheric and terminal pressures remaining as before. The compressor will still draw in 135 cubic feet of air per minute but, owing to the higher temperature, a lesser weight, or mass, of air will be withdrawn from the atmosphere. According to the law of Charles, previously given, 135 cubic feet of air at 65° F. and 14.7 pounds pressure is equivalent to 133.75 cubic feet of air at 60° F., and 14.7 pounds pressure. Under these conditions our expression for volumetric efficiency becomes

$$E_v = \frac{133.75}{150} = 89 \text{ per cent} \tag{77}$$

This shows that for a rise of every 5° F. in intake-air temperature there occurs a loss of practically 1 per cent in volumetric efficiency.

The values given in Table 20 show the effect of initial or intake temperature on the efficiency and capacity of air compressors. Unit capacity and efficiency is assumed to be at 60° F.

Likewise, volumetric efficiency is affected by change of atmosphere or intake pressure, the temperature remaining constant.

TABLE 20

Initial Temperature: Degrees Fahr.	Initial Temperature: Degrees Absolute	Relative Capacities and Efficiencies	Initial Temperature: Degrees Fahr	Initial Temperature: Degrees Absolute	Relative Capacities and Efficiencies
−20	441	1.18	70	531	.980
−10	451	1 155	80	541	.961
0	461	1.13	90	551	.944
10	471	1.104	100	561	.928
20	481	1.083	110	571	.912
30	491	1.061	120	581	.896
32	493	1.058	130	591	.880
40	501	1.040	140	601	.866
50	511	1 020	150	611	852
60	521	1 000	160	621	.838

To show this, let us suppose that our compressor were removed to a high altitude where intake air of 13.7 pounds pressure and 60° F. is available. Now, 135 cubic feet of this air is equivalent to 132.62 cubic feet of free air, and our third expression for volumetric efficiency is

$$Ev = \frac{132.62}{150} = 88.4 \text{ per cent} \qquad (78)$$

Therefore, for every 0.625 pound decrease of intake pressure there occurs a loss of 1 per cent in volumetric efficiency.

Formula and Measurement. — The most popular and convenient method of determining the volumetric efficiency of an air cylinder is from the indicator diagram. Referring to Fig. 104 the ratio $\frac{dc}{ce}$ is what we might call the *observed* volumetric efficiency. On the actual diagram, these distances are measured with some convenient scale, and the computations made and the results so obtained corrected for inlet temperatures and pressure. Volumetric efficiency found after these corrections are made is the *true* or *real* efficiency.

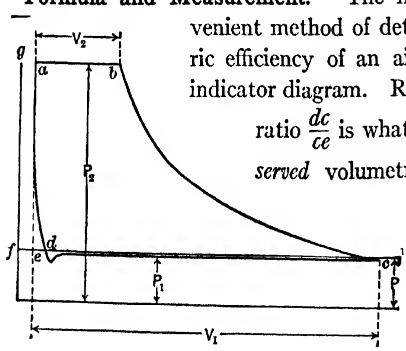


FIG. 104.

We may derive expressions for *real* and *observed* volumetric efficiencies from the diagram as follows:

Remembering that $PV^n = P_1V_1^n = P_2V_2^n = C$, from Fig. 104 we get:

$$\frac{df}{ag} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \qquad (79)$$

$$df = de + ag$$

Substituting in 79

$$\frac{de + ag}{ag} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

whence

$$de = ag\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \qquad (80)$$

ag is the clearance volume, and it may be determined by actual measurement by well-known methods. We may say, then, that this quantity is known, and, designating it as V_c and substituting in (79), we have

$$de = V_c\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \qquad (81)$$

Now

$$\textit{Observed } E_v = \frac{dc}{ce} = \frac{ce - de}{ce} \qquad (82)$$

where

$$ce = \text{piston displacement } P_d.$$

Substituting the value of de as given in (81) into (82) we get:

$$\textit{Observed } E_v = \frac{P_d - V_c\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]}{P_d} \qquad (83)$$

or

$$\textit{Observed } E_v = 1 - \frac{V_c}{P_d}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \qquad (84)$$

The ratio $\frac{V_c}{P_d}$ is, obviously, the percentage of the cylinder volume given up to clearance. Equation 84 shows that, in scaling the diagram for volumetric efficiency, we have taken into consideration the effect of clearance and terminal pressures but not that of the initial temperature and pressure. In order to provide for this, it is necessary to multiply 84 by $\frac{T_1}{T}$, where T is the absolute temperature of the air at the instant that compression begins, and T_1 is 60 + 460.6.

Also, the pressure at the beginning of compression is nearly always less than that of the atmosphere, due to frictional losses, valve spring resistance, etc. To provide for this, equation 84 must be multiplied by $\frac{P}{P_1}$, where P is absolute pressure shown by the intake line on the diagram, and P_1 is 14.7 pounds.

The expression for the true or real volumetric efficiency, therefore, is

$$True\ E_v = \frac{T_1 P}{T P_1}\left(1 - \frac{V_c}{P_d}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]\right) \qquad (85)$$

Under some circumstances this method of volumetric efficiency determination is to be avoided, for results then obtained will be misleading and, consequently, worse than worthless. Leaky suction valves or stuffing boxes, and a cylinder scored at or near the end of the stroke will produce an almost perpendicular re-expansion curve.

These defects may be detected on the diagram, however, by plotting the theoretical curves and comparing with the actual curves. Fig. 105 shows the typical case of leaky valves on one end and their effect on the volumetric efficiency and the compression curve. A better way to determine the volumetric efficiency of a compressor under all conditions of cylinder, etc., is to *actually measure* the air delivered and divide by the piston displacement. The air may be measured by means of a standard orifice or by a system of enclosed tanks. The former method is described by

Prof. Elmo G. Harris, in *Compressed Air*. The latter method is as follows:

Connect the air compressor to two enclosed tanks B and C, as in Fig. 106, with a regulating valve between B and C, air gauges

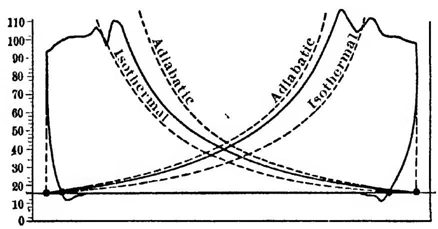


FIG. 105.

and thermometers as shown. By means of the regulating valve, the air pressure in B may be maintained at the desired pressure for which the volumetric efficiency is to be determined. The

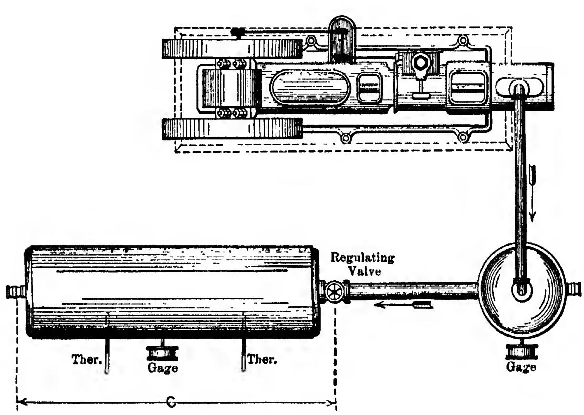


FIG. 106.

tank C then may be pumped up from pressure P_1 to P_2, both lower than the pressure in B.

It is always advisable to begin the test at an initial pressure higher than the atmospheric.

Now, shut off tank C and start the compressor. Watch the gauge on C until the needle reaches a suitable point, say 15 pounds, and from this time on until pressure P_2 is reached count the revolutions of the compressor, observe temperatures and time of run. A number of runs should be made and the mean of the results found substituted in the formula following.

Let P = atmospheric pressure − 14.7 pounds;
P_1 = initial absolute pressure in tank C;
P_2 = final absolute pressure in tank C;
T = absolute room temperature in degrees C.;
T_1 = absolute initial temperature of air in tank C;
T_2 = absolute final temperature of air in tank C;
V = volume in cubic feet of tank C;
V_1 = free air equivalent of air in tank at beginning of test;
V_2 = free air equivalent of air in tank at end of test;
v = actual amount of air pumped into tank;
$R_1 = \frac{P_1}{P}$ atmospheres at beginning of test;
$R_2 = \frac{P_2}{P}$ atmospheres at end of test;

Now, disregarding temperature,

$$v = V_2 - V_1 = \text{volume of air compressed or pumped.}$$

Now,

$$V_1 = V\frac{P_1}{P}; \text{ and } V_2 = V\frac{P_2}{P}$$

$$v = V\left(\frac{P_2}{P} - \frac{P_1}{P}\right)$$

or

$$v = V(R_2 - R_1) \tag{86}$$

The theoretical quantity of air pumped is equal to the piston displacement of the compressor, or $l \times a \times n$, where

l = length of stroke in feet;
a = area of piston in square feet;
n = number of strokes.

The volumetric efficiency is then represented by the formula

$$E_v = \frac{V\,(R_2 - R_1)}{l \times a \times n} \tag{87}$$

Correcting for temperatures, (86) becomes:

$$v = VT\left(\frac{R_2}{T_2} - \frac{R_1}{T_1}\right) \tag{88}$$

If the barometric pressure is other than 29.92 inches of mercury, the formula should be corrected and made to read as follows:

$$v = \frac{VT}{B}\left(\frac{29.92\,R_2}{T_2} - \frac{29.92\,R_1}{T_1}\right) \tag{89}$$

Our final expression for volumetric efficiency then becomes

$$Real\ E_v = \frac{\dfrac{29.92\,VT}{B}\left(\dfrac{R_2}{T_2} - \dfrac{R_1}{T_1}\right)}{lan} \tag{90}$$

A method used in Europe for accurately measuring the output of an air compressor is a system of air receivers somewhat similar to that just described, together with a low pressure air nozzle. The arrangement is illustrated in Fig. 107.

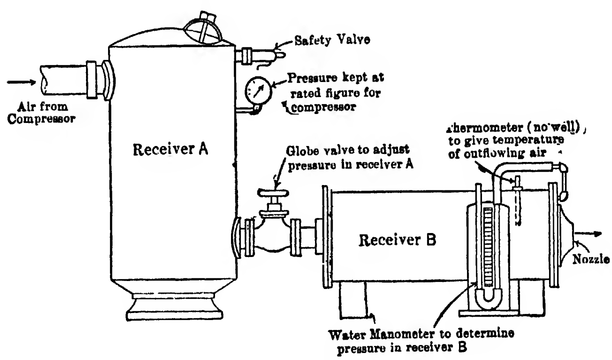


FIG. 107.

As shown, the air is delivered by the compressor into Receiver *A* where the pressure may be maintained or adjusted by manipulating the globe valve shown. Receiver *B* is provided at its end with a standard nozzle. A thermometer and a water manometer are also provided in Receiver *B*.

To insure absence of eddy currents, it is advisable that Receiver B be of ample proportions. The length should be at least 15 feet and the diameter not less than 3 times that of the nozzle.

Operation is as follows: Maintain the pressure in Receiver A at that point for which the volumetric efficiency of the compressor is desired. The nozzle in Receiver B should be of such size that a pressure of not over one pound gauge is maintained inside.

Make an accurate record of the number of revolutions of the compressor and take readings of the manometer and the thermometer, making the usual notations.

Remembering the relations existing between pressure volume and temperature and specific weights of air and water, the following formula may be readily derived:

$$V = 3.64\, Cd^2 \sqrt{\frac{HT}{P^m}} \tag{91}$$

where

V = cu. ft. of air per minute at observed temperature and atmospheric pressure;

d = smallest diameter of nozzle;

H = length of water column in inches;

P^m = absolute mean pressure in nozzle;

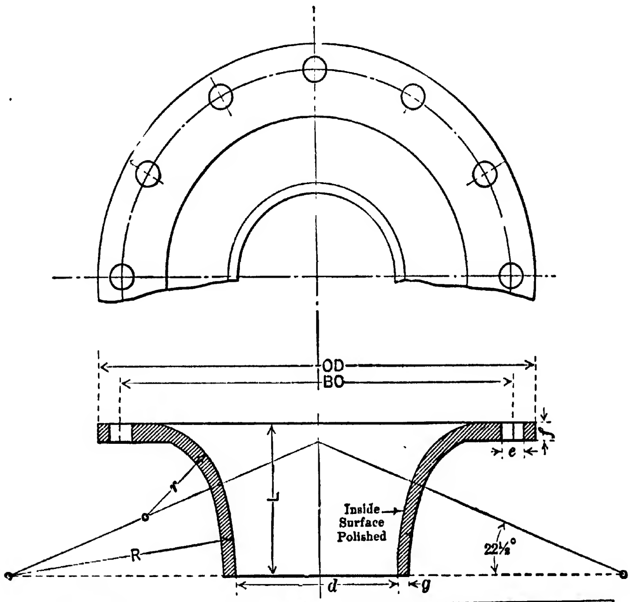
C = Nozzle coefficient = 0.98 to 0.99.

Obviously, the air flowing through the nozzle is usually quite hot, hence in computing the volumetric efficiency of a compressor, it is necessary to reduce the volume obtained from the above formula to an equivalent volume at 60° F. temperature. Assuming this to have been done, the expression for volumetric efficiency is

$$Real\ E_v = \frac{3.64\, Cd^2 \sqrt{\frac{HT}{P^m}}}{lan} \tag{92}$$

Figure 108 illustrates the standard form of nozzle which was adopted by the German Engineer-societies in 1912. A table of dimensions is also given.

Over-all Efficiency. — This efficiency is the most important of all to the user, for it refers directly without limitation or proviso to the cost of operation. It means the cost in fuel, up-



d″	L″	R″	r″	e″	OD	BC	NUMBER OF BOLTS	DRILLED FOR STD. FLANGE. SIZE	FREE AIR CU. FT. MAX. CAPACITY	FREE AIR CU. FT. MIN. CAPACITY	f″	g″
2	1.70	2.8	1.0	⅞	15:0	13¼	12	9″	340	140	⅝	½
3	2.55	4.2	1.5	⅞	15:0	13¼	12	9″	755	290	⅝	½
4	3.40	5.6	2.0	⅞	15:0	13¼	12	9″	1330	520	⅝	½
5	4.25	7.0	2.5	⅞	15:0	13¼	12	9″	2065	810	⅝	½
7	5.95	9.8	3.5	1	19:0	17.0	12	12″	3990	1560	¾	½
9	7.65	12.6	4.5	1⅛	23½	21¼	16	16″	6600	2575	¾	½
11	9.35	15.4	5.5	1¼	29½	27¼	20	22″	9860	3850	¾	½

Coefficient: 0.99 Capacities figured $\left\{\begin{matrix}\text{Max. } 30''\\ \text{Min. } 3''\end{matrix}\right\}$ Water Column

FIG. 108.

keep, supplies, interest and depreciation of air delivered to the receiver or pipe line, and is, consequently, a combined statement of mechanical, compression, and volumetric efficiencies as well as of reliability.

The following is a general expression of over-all efficiency referred to the isothermal, no clearance base.

$$E_0 = \frac{\text{Isothermal H.P. per 100 cubic feet of air per minute}}{\text{Boiler H.P. per 100 cubic feet per minute actually delivered}} \tag{93}$$

This expression does not take into consideration all the factors that affect over-all efficiency, but even as it is, it is something definite and a much more satisfactory guide than any yet given. Elaborate tests over long periods of time are necessary to determine the true over-all efficiency of any machine and, obviously, no accurate formula can be derived that will take into consideration even one of the variables, such as upkeep or depreciation. The most reliable information we have on the subject is a series of tests made by Mr. Richard L. Webb on a number of air compressors in the Canadian mining district. These tests are published in *Compressed Air Plant*, by Prof. Robert Peele.

Economy Essentials. — The foregoing discussion and statement of facts show that the economy of an air-compressor unit, depends:

(1) Upon the mechanical construction, that is, the size and proportion of bearings and wearing surfaces; lubricating system and general design of parts.

(2) Upon the length and volume of ports in the air cylinder. Long and tortuous ports and air passages increase the losses by heating the incoming air before compressions begins.

(3) Upon the cooling devices and water jackets and temperature of cooling water.

(4) Upon the surrounding conditions, that is, altitude at which the machine is being operated, and atmospheric temperature.

(5) Upon the clearance spaces.

(6) Upon the economy of the power end.

CHAPTER XI

THE COMPRESSOR

Air compressors as a whole are usually divided into two general classes, namely, *wet* and *dry*. The wet compressor is divided into three types, and the dry compressor is divided and subdivided into various types and designs until the commercial machine with its cylinder combinations and construction are reached. The diagram (Fig. 109) shows this progression quite clearly and will be found convenient in selecting a compressor to meet certain conditions and requirements.

The various makes and designs of air compressors have been illustrated and discussed most ably by Prof. Robert Peele in *Compressed Air Plant*, and compressor manufacturers issue catalogues and bulletins describing their product which may be had for the asking; hence nothing can be said here in this connection that would not be mere repetition.

Air-compressor Installation and Operation.* — The large majority of instances of unsatisfactory operation of air compressors, and often disastrous explosions in receivers and pipe lines emanate from improper installation in the first place and continued negligent operation and disregard of the compressor manufacturers' instructions in the second place. Unfortunately, many operating engineers look upon the compressor as a rough and ready machine built to withstand all manner of abuse, expensive to operate and only to be used when nothing else will serve the purpose. This impression is, of course, erroneous, and is easily corrected if the intended operator, before erecting a compressor, will familiarize himself with the practical principles of air compression; the attendant dangers and necessary precautions; and the simple requirements essential to economical

* *Power*, Dec. 30, 1913. *Air Compressor Installation and Operation*, by E. M. Ivens.

- Air compressors
  - Wet
    - Water injection type
    - Hydraulic type
    - Taylor type
  - Dry
    - Rotary type
    - Turbine type
    - Positive blower type
    - Reciprocating type
      - Gas, gasoline and oil engine driven
      - Steam driven
        - Straight line
          - Simple steam, single-stage air;
          - Simple steam, two- and three-stage air;
          - Compound steam, single-stage air;
          - Compound steam, two-stage air.
        - Duplex
          - Duplex steam, single-stage air;
          - Duplex steam, two-, three-, four-stage air;
          - Compound steam, single-stage air;
          - Compound steam, two-stage air;
          - Compound steam, three-stage air;
          - Compound steam, four-stage air.
        - Triplex
          - Triple-expansion steam single-stage air;
          - Triple-expansion steam, two-stage air;
          - Triple-expansion steam, three-stage air.
        - Quadruplex
          - Quadruple-exp. steam, single-stage air;
          - Quadruple-exp. steam, two-stage air;
      - Power driven
        - Straight line
          - Single-, two-, three- and four-stage belted;
          - Single-, two-, three- and four-stage geared;
          - Single-, two-, three- and four-stage chain.
        - Duplex
          - Single-, two-, three- and four-stage belted;
          - Single-, two-, three- and four-stage geared;
          - Single-, two-, three- and four-stage chain;
          - Single-, two-, three- and four-stage direct connected.

FIG. 109.

operation. A few instructions follow which, if observed only indifferently, will furnish the much needed relief to the compressor as well as reduce operation costs and eliminate the danger of the now too frequent explosions.

Location. — In installing a new compressor the first consideration to come up is where to build the foundation. The cleanest and coolest place available in the room should be chosen and ample space should be provided all around for cleaning and inspecting the compressor. Location in boiler rooms and near coal piles should be especially avoided.

Foundation. — The size and depth of foundation depends upon the size and type of compressor, and upon the nature of the soil. With each compressor the manufacturers send out a detailed foundation plan, assuming that the foundation will be built in firm ground. If, however, the ground is insecure in any way, a liberal base, a foot or more larger all around than the bottom of the foundation, should be added to the manufacturers' specifications.

Owing to the nature of the work a compressor has to perform, there are certain shocks, with the resulting vibration, that must be absorbed by the foundation. It is always advisable in building foundations for straight-line machines, that special pains be taken to make them of liberal size and rigidity. It is a good plan to reënforce concrete foundations with $\frac{1}{2}$- to $\frac{5}{8}$-inch iron rods near the top and bottom, placing some lengthwise, and others crosswise of the foundation. In the duplex-type compressors, the unbalanced strains are somewhat eliminated by the quartering-crank arrangement, but a good foundation costs but little more in the first place and is always to be desired.

The material for foundations may be burned brick, stone or cement concrete. If either of the first two is chosen, thin and well-grouted joints of cement mortar of one part Portland cement to two parts of sharp sand, should be made. If concrete is used, a mixture of one part Portland cement, three parts of sharp sand and five parts of crushed stone or gravel will be found quite satisfactory.

Receiver. — The functions of the air receiver are (1) to create a cushion and thereby eliminate the compressor pulsations in the pipe line; (2) to serve as a storage of power; (3) to cool the air and precipitate any oil or moisture carried in entrainment; (4) to eliminate certain friction losses that would occur if cooling were effected in the pipe lines. The receiver should, consequently, be located in a cool place, preferably outside of the building, and as close as possible to the compressor.

Receiver fittings should include pressure gauge, safety valve and blow-off cock located at or near the bottom.

Air-inlet Piping. — It has already been shown that an increase of 5° F. in temperature of intake air is accompanied by a decrease of 1 per cent in volumetric efficiency; which means that as the intake-air temperature increases, the free-air capacity of the machine decreases and the same amount of energy is expended as though the full capacity of the machine were being realized. To assist the compressor, then, the inlet should be piped to the outside of the building and some ten or twelve feet above the ground surface. The opening should be well screened to prevent drawing in dust and dirt, and hooded to keep out rain. If it is impracticable to carry the intake outside and air must be drawn into the cylinder directly from the room, it is very important that no dust or dirt be allowed near the opening, for a small amount of dirt being continually drawn into the cylinder with the air will cut and wear the inner surfaces and valves very rapidly, and no end of trouble results.

Sometimes conduits are used instead of piping to the inlet. These are best constructed of wood lined with tin and the opening well screened.. Concrete or brick construction should be avoided, for grit is likely to be loosened by the vibrations of the compressor and drawn into the cylinder. Conduits should be at least double the cross-sectional area of the inlet opening of the compressor.

As few bends as possible should be put in the inlet piping and when used, should be either long turn fittings or pipe neatly bent, preferably the latter. To further reduce frictional resistance,

the inlet piping should be increased in diameter in proportion to its length. A good rule to follow is to increase the diameter one-half inch for each ten feet added in length.

Discharge Piping. — The pipe connecting the compressor and the receiver should at least be of the diameter of the discharge opening of the cylinder and contain as few bends as possible. Very often a salesman, in taking an order for an air receiver, recommends one whose inlet opening is considerably smaller than the compressor discharge opening. When the receiver arrives and the engineer on the ground learns this, he immediately proceeds to insert a pyramid of bushings in the cylinder opening. This imposes additional hardship on the compressor and creates a condition conducive to explosion as we shall see later.

Another serious mistake often made is the placing of a stop valve between the compressor and the receiver. This should never be done unless a safety valve be placed between the stop valve and the air cylinder; for there is a possibility at some time of starting up with the stop valve closed, when dangerous pressure will soon be reached and explosion likely to occur.

Lubrication. — The bearings and other external wearing parts of the air compressor are usually lubricated either by means of oil and grease cups suitably placed or by the splash or bath system. The latter method is coming into more popular favor and is rapidly replacing the former because of its simplicity, effectiveness and economy in the use of oil. Against it stands the objection that it is likely to be neglected and the oil becomes dirty and gritty, due to accummulation of abrasives gathered by the oil in passing and repassing over the bearings.

The air-cylinder lubrication is by far the most vital point in air-compressor operation, and it seems to be the least understood. In order to appreciate fully the necessity of proper cylinder lubrication, consider the conditions that have to be met.

The compression of a gas is accompanied by a rise in temperature, in accordance with the law stated in Chapter IX. For

the adiabatic compression of air, the temperature and pressure relations are expressed by the formula:

$$\frac{T_1}{T} = \left(\frac{P_1}{P}\right)^{\frac{n-1}{n}} = \left(\frac{P_1}{P}\right)^{.29}$$

whence

$$T_1 = T\left(\frac{P_1}{P}\right)^{.29} \qquad (94)$$

where T and T_1 are the initial and the final absolute air temperatures respectively, and P and P_1 the initial and the final absolute pressures. Therefore, the temperature of the air at discharge from the cylinder is dependent not only upon the pressure but upon the temperature of the intake air. Suppose that we now have a single stage compressor operating at sea level and that the atmospheric temperature is 60° F., and the discharge pressure 70 pounds, the final temperature is

$$T_1 = 521^\circ\left(\frac{84.7}{14.7}\right)^{.29} = 866^\circ \text{ absolute}$$

or 406 degrees by the thermometer. This calculation is based upon no heat radiation losses, and is, consequently, slightly greater than the actual discharge temperature.

The difference between actual discharge temperatures and that calculated above is small, for the actual compression line follows the adiabatic very closely. Air is one of the poorest conductors of heat, and the water jacket has little effect other than to facilitate lubrication. Tests of compressors operating under conditions named show that the actual discharge air temperatures range between 325° and 365° F., and instances even of higher temperatures are on record.

Table 21 gives the temperatures of air when compressed adiabatically from atmospheric pressure and 60° F., up to the various gauge pressures indicated in column 1. The calculations were made by substituting in formula 92.

TABLE 21

I.	2.		
Guage Pressure	Absolute Pressure	Pressure in Atmospheres	Temperatuers Air not cooled
0	14 7	1.	60
5	19.7	1.34	106
10	24.7	1.68	145
15	29.7	2 02	178
20	34 7	2.36	207
25	39.7	2.70	234
30	44 7	3.04	255
35	49.7	3 38	281
40	54.7	3 72	302
45	59.7	4 06	321
50	64.7	4 40	339
55	69.7	4.74	357
60	74.7	5.08	375
65	79.7	5 42	389
70	84 7	5 76	405
75	89 7	6 10	420
80	94 7	6 44	432
85	99 7	6 78	447
90	104.7	7 12	459
95	109.7	7.46	472
100	114.7	7.80	485
105	119 7	8 14	496
110	124.7	8 48	507
115	129 7	8.82	518
120	134 7	9 16	529
125	139 7	9 50	540
130	144 7	9.84	550
135	149 7	10.18	560
140	154 7	10 52	570
145	159.7	10 85	580
150	164.7	11 20	589

The lowest temperature at which an oil will give off combustible vapors is called the *flash point* and the temperature at which these vapors ignite and continue to burn is called the *ignition point.* The flash point of common lubricating oil is about 260° F., and the ignition point about 295 degrees. Common cylinder oils

flash at about 350 degrees and ignite at about 400° F. The oil best suitable to air cylinder service is one having a flash point of about 500 degrees and an ignition point of about 600° F.

If proper oil is used, a comparison of temperatures will show that, under ordinary conditions and with cylinder and valves in good condition, an explosion is impossible. If, on the other hand, a low-flash-test cylinder oil is used, it is soon decomposed by the heat, the volatile constituents ignited and a destructive explosion usually follows. There are instances on record where ignition has occurred without explosion, but the chances are always in favor of explosion.

A scored cylinder and valves, caused by dirt and grit drawn in with the air and sticking discharge valves, may also cause the ignition of volatile constituents of the oil. For instance, suppose that a sufficient amount of the air at discharge temperature to raise the initial temperature from 60° F. to 200° F. found its way back from the receiver or pipe line into the cylinder on the suction side of the piston. This air might return through either leaky or sticking discharge valves, or from the compressing side of the moving piston to the suction side. Then with 200° F. initial temperature, the final temperature of air compressed to 70 pounds gauge would be

$$T_1 = 661\left(\frac{84.7}{14.7}\right)^{.29} = 1096 \text{ degrees absolute,}$$

or 635 degrees by the thermometer, which is high enough to decompose and ignite even the best of oils. This shows the importance of locating the compressor so that the coolest and cleanest air obtainable is drawn into the cylinder. Other conditions favorable to ignition are carbon deposits from the oil on the valves and passages, restricting their area; too small a discharge pipe; and drawing air from a hot engine room. Any of these cause at least an increased final temperature, and each, if extreme, will ultimately cause ignition.

Figure 110 shows the Hodges' fusible alarm plug, manufactured by the Ingersoll-Rand Co. By simply drilling and tapping a

$\frac{1}{4}$-inch hole, this little device may be installed in any pipe line. In case of an abnormally high temperature the fusible insert of the plug melts and the air escapes with a sharp whistle. This immediately attracts the attention of the operator, and the cause for the high temperature may be remedied before any serious damage is done.

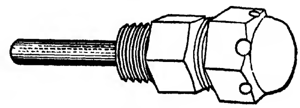


FIG. 110.

Only a very small amount of oil is necessary for the air cylinder and as little as possible should be used, for excess of oil will deposit carbon and gum the valves. Just how much can best be determined by experiment, but a good approximation is one drop per minute for cylinders from 6- to 10-inch stroke, three drops in two minutes for cylinders from 12- to 16-inch stroke, two drops per minute on 18- to 24-inch stroke cylinders, and three to five drops for larger cylinders. These quantities apply when the compressor is running at normal speed, and if, for any reason, the revolutions are increased or decreased, the quantity of oil should also be varied in proportion.

Circulating Water. — The duty of the jacket water is to carry off the heat transmitted to the cylinder walls and heads by the compression of the air, and thereby assist lubrication. A liberal supply of cool water should be furnished the jacket and necessary precautions taken that will prevent starting up with a dry jacket.

Air cylinders are provided with water inlet and outlet openings as well as drain. In some cylinders, inlet and outlet openings are at the top of the barrel, while in others, the inlet is below and the outlet above the barrel. In the first instance, there can be no mistake in making pipe connections, but with the latter arrangement of openings, the error is often made of connecting the inlet pipe above into the outlet opening. When this is done, the jacket is not kept full of water, and the surfaces not in contact with the water will become heated.

The water outlet should be in plain view of the operator, and this is best accomplished by allowing the water to fall into an

open pipe end or funnel, as shown in Fig. 111. The controlling valve should always be placed in the inlet. Sometimes the circulating water is used for other purposes after leaving the jacket, and a closed circuit is necessary. The jacket water pressure should not exceed 50 to 60 pounds unless special attention has been given to the design. Dirty circulating water is injurious in that mud deposits form which prevent the water from reaching the metal and heating will result. If the compressor is exposed to freezing temperature, the jacket should be drained after being shut down, otherwise the expansion of the water in freezing will crack the jacket.

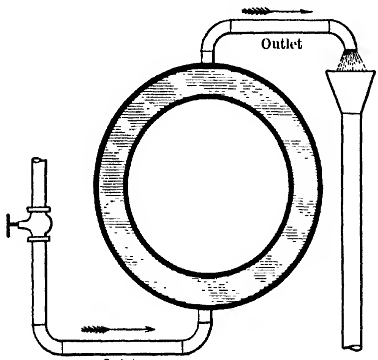


FIG. 111.

Inspection and Cleaning. — At stated intervals, say every month, the compressor should be thoroughly inspected and any defect immediately corrected. Usually, the air valves of the modern compressor are placed conveniently and can be easily removed and examined. They should present an oily surface and be kept free of carbonaceous deposits. The ports and passages should also be kept clean and free from obstructions.

Cleaning the inside of the air cylinder may be done effectively by filling the lubricator with a strong solution of water and soap, and feeding liberally throughout a day's run. Generous quantities are necessary, because soap in itself is not a very good lubricant. At the end of the day's run the lubricator should be filled with oil and the compressor operated for awhile; this, to prevent rusting of the inner polished surfaces. A soap-sud lubricator suggested by Mr. Martin McGerry in *Power*, is shown in Fig. 112. *A* is a galvanized water tank and *B* is a

smaller tank soldered to the side of A and containing soap. The bottom of B is perforated with $\frac{1}{8}$-inch holes as shown at C. The water in A passes through the holes up into B and in passing dissolves some of the soap and rises to the top of the $\frac{1}{4}$-inch pipe E. The solution then passes down this pipe into the compressor suction. Before shutting the compressor down, the water supply at D may be turned off and oil from the cup G fed into B.

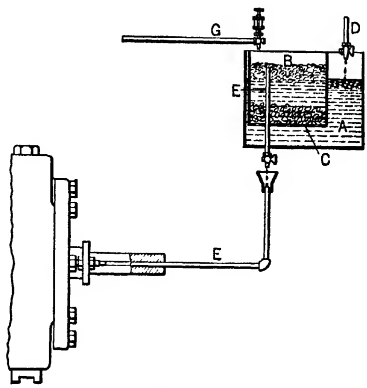


FIG. 112.

Because of its low flash point, kerosene should never be used for cleaning the cylinder. Summing up all that has been said, the following mode of operation is obviously to be recommended:

Every morning: —

1. Drain the receiver;
2. Note the height of lubricating oil in the crank case (or in oil cups) and replenish if necessary;
3. Adjust lubricator for proper amount of oil feed;
4. Start circulating water.

Every week: —

5. Remove crank case, oil and filter;
6. Remove and examine suction and discharge valves. If worn or cut, they should be ground to a tight fit;
7. Test the safety valve by raising the air pressure to the point of blow-off;
8. Take up lost motion in pins and bearings.

Every month: —

9. Renew crank case oil, and throughly cleanse the inside of crank case;
10. Thoroughly inspect all parts, including air and water passages.

CHAPTER XII

FLOW OF COMPRESSED AIR IN PIPES

As we shall have to do with the flow of both air and water in pipes, it is well that we review the principles and laws governing each. There is more or less approximation in all the calculations and formulæ for pressure loss in water and air transmission and, owing to some uncontrollable variables involved, it is probable that truly accurate formulæ will never be had until a great deal more experimental data is available.

'Water is practically incompressible, and is of approximate constant density under all ordinary pressures. Consequently, the water frictional losses in pipe are independent of pressure conditions and the losses in any given section of a pipe line of uniform diameter are identical with those occurring in any other section of the same diameter, length and character. In other words, loss of head due to friction is directly proportional to the length of pipe through which the water flows.

Air, on the other hand, is very elastic and the volume is inversely proportional to the absolute pressure exerted upon it. Compressed air then advancing in a pipe line encounters a head or pressure loss due to friction and expands in proportion. The velocity of flow is increased in consequence, and this reacts to further increase friction loss and so on. The air friction loss, then, unlike water friction loss, varies in each unit of distance in a pipe line, and herein lies one difficulty of accurate calculation. The compressed air friction tables in general use at this writing are based on the assumption that air friction loss is directly proportional to the length of pipe; that is, if a certain loss occurs in 1000 feet of pipe, the loss in 2000 feet will be twice as great. This means uniform velocity throughout the length of the line and in order to realize which, it would be necessary to con-

struct the pipe line of gradually increasing cross-sectional area. True, the friction loss of air increases with the length of piping, the number of elbows, and so forth; but the loss in each unit of length added is greater than that in the preceding one, and the loss in the last unit is considerably greater than that in the first one. Consequently, the shorter the air line, the more nearly correct will be the tables generally used; and the longer the line, the greater will be the discrepancy.

Another fallacious assumption sometimes made is that regarding the relation of friction loss to diameter of pipe. The interior resistance is much greater in proportion to volume transmitted in small pipes than in large ones because as the diameter is reduced, the ratio of perimeter to cross-sectional area increases. In forcing a given volume of compressed air through a 1-inch line the loss is about $3\frac{1}{4}$ times that encountered in forcing an equal volume through a 2-inch line of the same length. Other incalculable variables affecting friction are irregularities on the inner surfaces of the pipe, and the broken surface at each joint.

To sum up, the laws of air friction are:

1. The loss of pressure due to friction increases with the length of pipe;

2. It increases with the square of the volume of air being transmitted;

3. It increases with the roughness of the interior surface of the pipe;

4. It increases with the number of bends, joints and fittings;

5. It increases as the diameter of the pipe is reduced.

These laws are expressed by D'Arcy in his formula:

$$Q = C\sqrt{\frac{d^5(P_1 - P_2)}{wl}} \tag{95}$$

$$= \frac{C\sqrt{d^5}}{\sqrt{l}}\sqrt{\frac{P_1 - P_2}{w}}$$

whence

$$P_1 - P_2 = \frac{wQ^2l}{C^2d^5} \tag{96}$$

where

P_1 = initial gauge pressure at receiver;

P_2 = final gauge pressure at the end of pipe line;

$P_1 - P_2$ = pounds pressure loss in friction;

w = weight of air in pounds per cubic foot at pressure P_1;

Q = volume of compressed air delivered in cubic feet per minute,

l = length of pipe in feet;

C = experimental coefficient depending upon pipe diameter;

d = diameter of pipe in inches.

In Table 22 may be found the values of w under varying temperatures and pressures, and in Table 23* are given values of C, d^5 and $C\sqrt{d^5}$ for various pipe diameters up to and including 12 inches.

Mr. Nathaniel Herz has solved D'Arcy's formula graphically in the December, 1912, Bulletin of the A. I. M. E. He explains his chart which is reproduced in Fig. 113 as follows:

"The most common case is that in which the given quantities are: the quantity of air required, the length of the pipe, and the initial pressure. The method of solution is to assume a pressure loss and to compute the remaining factor, thus giving the size of pipe corresponding to the assumed loss of pressure. It is always desirable to try two or more pressure drops, in order to find the combination that is most satisfactory, since often a small change in the size of pipe will reduce or increase the loss of pressure several pounds. An alternative method is to assume a size of pipe and calculate the corresponding pressure drop. Each method involves a series of tedious calculations to arrive at the most economical solution, and also requires the use of tables giving the constant, c, the actual diameters corresponding to the nominal pipe sizes, the density of the air, and often for convenience, a table giving the value of expression. A graphic chart has been constructed for the

* Robt. Peele — *Compressed Air Plant.*

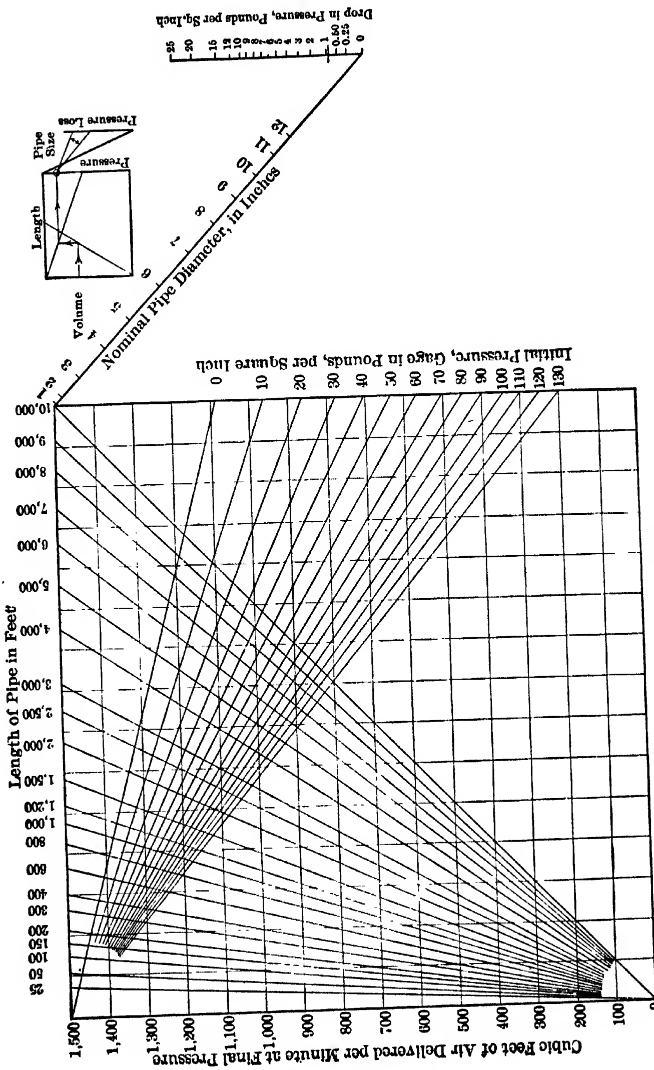

Drop in Pressure, Pounds per Sq. Inch
Nominal Pipe Diameter, in Inches
Pipe Size
Pressure Loss
Pressure
Length
Volume
Initial Pressure, Gage in Pounds, per Square Inch
Length of Pipe in Feet
Cubic Feet of Air Delivered per Minute at Final Pressure

FIG. 113.

TABLE 22

WEIGHT OF AIR AT VARIOUS PRESSURES AND TEMPERATURES (Based on an Atmospheric Pressure of 14.7 Pounds Absolute at Sea Level)

Temp. of air, deg. Fahr.	Gauge pressure, pounds																				
	0	5	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250	300
	Weight in pounds per cubic foot																					
−20	0.0900	0.1205	0.1515	0.2125	0.2744	0.3360	0.3970	0.4580	0.5190	0.5800	0.6410	0.702	0.7635	0.825	0.886	0.948	1.010	1.165	1.318	1.465	1.625	1.930
−10	0.0882	0.1184	0.1485	0.2090	0.2685	0.3283	0.3880	0.4478	0.5076	0.5674	0.6272	0.687	0.747	0.807	0.868	0.928	0.989	1.139	1.288	1.438	1.588	1.890
0	0.0864	0.1160	0.1455	0.2040	0.2630	0.3215	0.3800	0.4385	0.4970	0.5555	0.6140	0.672	0.731	0.790	0.849	0.908	0.968	1.114	1.260	1.406	1.553	1.850
10	0.0846	0.1136	0.1425	0.1995	0.2568	0.3145	0.3720	0.4292	0.4863	0.5433	0.6006	0.658	0.716	0.774	0.832	0.889	0.947	1.090	1.233	1.376	1.520	1.810
20	0.0828	0.1112	0.1395	0.1955	0.2516	0.3071	0.3645	0.4205	0.4770	0.5330	0.5890	0.645	0.701	0.757	0.813	0.869	0.927	1.067	1.208	1.348	1.489	1.770
30	0.0811	0.1088	0.1366	0.1916	0.2465	0.3015	0.3570	0.4121	0.4672	0.5221	0.5771	0.632	0.687	0.742	0.797	0.852	0.908	1.046	1.184	1.322	1.460	1.735
40	0.0795	0.1067	0.1338	0.1876	0.2415	0.2954	0.3503	0.4038	0.4576	0.5114	0.5652	0.619	0.673	0.727	0.781	0.835	0.890	1.025	1.161	1.296	1.431	1.701
50	0.0780	0.1045	0.1310	0.1839	0.2367	0.2905	0.3432	0.3960	0.4487	0.5014	0.5541	0.607	0.660	0.713	0.766	0.819	0.873	1.006	1.139	1.271	1.403	1.668
60	0.0764	0.1025	0.1283	0.1803	0.2323	0.2840	0.3562	0.3882	0.4402	0.4927	0.5447	0.596	0.649	0.700	0.752	0.804	0.856	0.988	1.116	1.245	1.376	1.636
70	0.0750	0.1005	0.1260	0.1770	0.2280	0.2791	0.3302	0.3808	0.4316	0.4824	0.5332	0.584	0.635	0.686	0.737	0.788	0.839	0.967	1.095	1.223	1.350	1.604
80	0.0736	0.0988	0.1239	0.1738	0.2237	0.2739	0.3242	0.3738	0.4234	0.4729	0.5224	0.572	0.622	0.673	0.723	0.774	0.824	0.949	1.074	1.199	1.325	1.573
90	0.0723	0.0970	0.1218	0.1707	0.2195	0.2688	0.3182	0.3670	0.4154	0.4639	0.5122	0.561	0.611	0.660	0.709	0.759	0.809	0.932	1.054	1.177	1.300	1.544
100	0.0710	0.0954	0.1197	0.1676	0.2155	0.2638	0.3122	0.3602	0.4079	0.4555	0.5033	0.551	0.599	0.648	0.696	0.745	0.794	0.914	1.035	1.155	1.276	1.517
110	0.0698	0.0937	0.1176	0.1645	0.2115	0.2593	0.3070	0.3542	0.4011	0.4481	0.4950	0.542	0.589	0.637	0.685	0.732	0.780	0.899	1.017	1.135	1.254	1.491
120	0.0686	0.0921	0.1155	0.1618	0.2080	0.2549	0.3018	0.3481	0.3944	0.4403	0.4866	0.533	0.579	0.626	0.673	0.720	0.767	0.884	1.001	1.118	1.234	1.465
130	0.0674	0.0905	0.1135	0.1590	0.2045	0.2505	0.2966	0.3446	0.3924	0.4296	0.4770	0.524	0.570	0.616	0.662	0.708	0.754	0.869	0.984	1.099	1.214	1.440
140	0.0663	0.0889	0.1115	0.1565	0.2015	0.2465	0.2915	0.3364	0.3813	0.4262	0.4711	0.516	0.561	0.606	0.651	0.696	0.742	0.855	0.968	1.081	1.194	1.416
150	0.0652	0.0874	0.1096	0.1541	0.1985	0.2425	0.2865	0.3308	0.3751	0.4193	0.4636	0.508	0.552	0.596	0.640	0.685	0.730	0.841	0.953	1.064	1.175	1.392
175	0.0626	0.0840	0.1054	0.1482	0.1910	0.2335	0.2755	0.3181	0.3607	0.4033	0.4450	0.488	0.531	0.573	0.616	0.658	0.701	0.808	0.914	1.021	1.128	1.337
200	0.0603	0.0809	0.1014	0.1427	0.1840	0.2248	0.2655	0.3054	0.3473	0.3882	0.4291	0.470	0.511	0.552	0.592	0.633	0.674	0.776	0.879	0.982	1.084	1.287
225	0.0581	0.0779	0.0976	0.1373	0.1770	0.2163	0.2555	0.2949	0.3344	0.3738	0.4129	0.452	0.491	0.531	0.570	0.609	0.649	0.747	0.846	0.944	1.043	1.240
250	0.0560	0.0751	0.0941	0.1323	0.1705	0.2085	0.2466	0.2845	0.3223	0.3602	0.3981	0.436	0.474	0.513	0.551	0.589	0.627	0.722	0.817	0.912	1.007	1.197
275	0.0541	0.0726	0.0910	0.1278	0.1645	0.2011	0.2378	0.2745	0.3111	0.3478	0.3844	0.421	0.458	0.494	0.531	0.568	0.605	0.697	0.789	0.881	0.972	1.155
300	0.0523	0.0707	0.0881	0.1237	0.1592	0.1945	0.2300	0.2654	0.3008	0.3362	0.3716	0.407	0.442	0.478	0.513	0.549	0.585	0.673	0.762	0.852	0.940	1.118
350	0.0491	0.0658	0.0825	0.1160	0.1495	0.1828	0.2160	0.2492	0.2824	0.3156	0.3488	0.382	0.415	0.449	0.482	0.516	0.549	0.632	0.715	0.799	0.883	1.048
400	0.0463	0.0621	0.0779	0.1090	0.1405	0.1720	0.2035	0.2348	0.2661	0.2974	0.3287	0.360	0.391	0.423	0.454	0.486	0.517	0.596	0.674	0.753	0.831	0.987
450	0.0437	0.0586	0.0735	0.1033	0.1330	0.1628	0.1925	0.2220	0.2515	0.2810	0.3105	0.340	0.369	0.399	0.429	0.458	0.488	0.562	0.637	0.711	0.786	0.934
500	0.0414	0.0555	0.0696	0.0978	0.1260	0.1540	0.1820	0.2100	0.2380	0.2660	0.2940	0.322	0.351	0.379	0.407	0.435	0.463	0.534	0.604	0.675	0.746	0.885
550	0.0394	0.0528	0.0661	0.0930	0.1198	0.1464	0.1730	0.1996	0.2262	0.2528	0.2794	0.306	0.333	0.359	0.386	0.413	0.440	0.507	0.573	0.641	0.709	0.841
600	0.0376	0.0504	0.0631	0.0885	0.1140	0.1395	0.1650	0.1904	0.2158	0.2412	0.2668	0.292	0.317	0.343	0.368	0.393	0.419	0.483	0.547	0.611	0.675	0.801

TABLE 23

Diameter of pipe (inches)	Values of C	Fifth powers of d	Values of $c\sqrt{d^5}$
1	45.3	1	45.3
2	52.6	32	297
3	56.5	243	876
4	58.0	1,024	1,856
5	59.0	3,125	3,298
6	59.8	7,776	5,273
7	60.3	16,807	7,817
8	60.7	32,768	10,988
9	61.0	59,049	14,812
10	61.2	100,000	19,480
11	61.8	161,051	24,800
12	62.0	248,832	30,926

solution of these problems with no computation, and without the use of tables. The procedure is as follows: — Begin with the quantity of compressed air delivered, on the left-hand vertical scale; follow across horizontally to the intersection with the inclined line corresponding to the length of the pipe line; pass up vertically to the inclined line corresponding to the initial pressure; then cross the chart horizontally to the heavy line at the right of the cross-sectioned part of the chart. The point here found is a pivot point, which is held with a pencil, pen, or needle point, and a straight-edge placed against it and swung across the "*Z*" diagram. Any two points on the inclined and vertical lines that are cut by the straight-edge at the same time go together as one solution of the problem, giving a pipe diameter with its corresponding loss of pressure. By swinging the straight edge, it is possible to see at a glance how the final pressure is effected by a variation of 1 inch in the pipe size. Moreover, the size giving the most desirable result is determined at one operation. If the drop is considerable, it may be desirable to adjust the volume to correspond with the new final pressure, and to repeat the operation; but within ordinary economical limits, the error involved by not doing so is negligible. Sometimes the problem may arise in another form; for instance, to find the maximum volume that can be handled in an existing line. In this case, the process is reversed. Begin with the maximum desirable drop, and the size of pipe, then pass to the initial pressure line in a horizontal direction, then vertically to the length line, and finally horizontal to the left-hand scale, which will give the corresponding volume. Any other combination can be solved in a similar manner. The accuracy of this chart is well within commercial limits. It has been checked against calculated values for combinations varying from 100 to 1000 cubic feet of compressed air delivered per minute, pressure losses from three to 10 pounds and pipes from 10 to 4000 feet long; all results were within 0.5 inches of the pipe diameter, and most of them within 0.25 inch or less."

TABLE 24
Loss of Pressure Caused by Friction of Compressed Air in Pipes

Equivalent volume in cubic feet of free air per minute passing through pipe	Size of pipe																		
	1	1¼	1½	2	2½	3	3½	4	5	6	7	8	9	10	12	14	16	18	20
	$P_1^2 - P_2^2$ = difference in squares of initial and final absolute pressures, per 100 feet of pipe																		
50	150	49.2	20	4.8															
75	335	98.5	46.2	10.5															
100	600	197	79	18.8	6.15														
150	1,350	443	178	42.2	13.8	5.5													
200	2,400	788	316	75.0	24.6	9.9	2.0	2.4											
250	3,750	1,230	494	117	38.5	15.4	7.1	3.6											
300	5,400	1,770	711	168	55.4	22.2	10.3	5.3											
400	9,600	3,150	1,263	300	98.3	39.5	18.2	9.4	3.1										
500	15,000	4,920	1,978	470	154	62	28.5	14.6	4.8										
600	21,600	7,100	2,845	675	221	89	41	21	6.9										
800	.....	12,400	5,070	1,200	394	154	73	37.5	12.3	4.94									
1,000	.....	19,600	7,900	1,880	615	247	114	58.4	19.2	7.7	2.75								
1,500	.....	44,000	17,800	4,220	1,380	555	255	132	43.2	17.4	8.0								
2,000	.....	.....	31,600	7,500	2,460	990	455	234	77.0	30.9	14.2	7.3							
3,000	.....	.....	71,000	16,800	5,540	2,220	1,030	526	173.0	69.5	32.1	16.5	9.15						
4,000	.....	.....		30,000	9,830	3,950	1,820	937	307.5	124	57	29.3	16.2	9.6					
5,000	.....	.....		47,000	15,500	6,180	2,850	1,460	480.0	193	89	45.7	25.4	15.0	6.03				
6,000	.....	.....		68,000	22,300	8,900	4,100	2,100	691.0	278	128	66	36.5	21.6	8.7				
8,000	.....	.....			30,600	15,800	7,300	3,750	1,230	494	228	117	65.5	38.5	15.4	7.1			
10,000	.....	.....			62,000	24,600	11,400	5,850	1,920	771	356	183	101.8	60	24.1	11.2	5.7		
15,000	.....	.....				55,000	25,500	13,180	4,320	1,740	801	412	228	135	54.1	25.0	12.9	7.1	
20,000	.....	.....				98,000	45,500	23,500	7,690	3,090	1,425	731	406	240	96.5	44.5	23.0	12.7	7.5
25,000	.....	.....					71,000	36,500	12,000	4,950	2,200	1,130	635	375	150	70	36.0	19.8	11.7
30,000	.....	.....						53,000	17,300	6,950	3,200	1,650	920	540	217	100	51.5	28.5	16.8
35,000	.....	.....						72,000	23,500	9,500	4,400	2,250	1,250	735	295	137	70.0	39.0	23
40,000	.....	.....						93,500	30,700	12,400	5,700	2,950	1,630	960	386	178	91.5	50.5	30
50,000	.....	.....							48,000	19,300	8,900	4,600	2,540	1,500	603	280	143	79.5	47
60,000	.....	.....							71,000	27,800	12,800	6,600	3,660	2,160	870	414	206	114	67
80,000	.....	.....								49,500	22,800	11,800	6,500	3,950	1,540	715	366	204	120
100,000	.....	.....								77,100	35,600	18,400	10,200	6,000	2,420	1,120	572	317	188

TABLE 25
Pressures and Squares of Pressures

Gauge pressure	Absolute pressure	Square of absolute pressure	Gauge pressure	Absolute pressure	Square of absolute pressure	Gauge pressure	Absolute pressure	Square of absolute pressure	Gauge pressure	Absolute pressure	Square of absolute pressure
0	14.7	216	......	......	......	......	......	......	......	......	......
2	16.7	279	......	......	......	......	......	......	......	......	......
4	18.7	350	......	......	......	......	......	......	......	......	......
6	20.7	428	......	......	......	......	......	......	......	......	......
8	22.7	515	......	......	......	......	......	......	......	......	......
10	24.7	610	56	70.7	4,998	105	119.7	14,328	240	254.7	64,855
12	26.7	713	58	72.7	5,285	110	124.7	15,550	250	264.7	70,055
14	28.7	824	60	74.7	5,580	115	129.7	16,822	260	274.7	75,450
16	30.7	942	62	76.7	5,883	120	134.7	18,144	270	284.7	81,050
18	32.7	1069	64	78.7	6,194	125	139.7	19,516	280	294.7	86,845
20	34.7	1204	66	80.7	6,512	130	144.7	20,938	290	304.7	92,840
22	36.7	1347	68	82.7	6,839	135	149.7	22,410	300	314.7	99,040
24	38.7	1498	70	84.7	7,174	140	154.7	23,932	325	339.7	115,400
26	40.7	1656	72	86.7	7,517	145	159.7	25,504	350	364.7	132,940
28	42.7	1823	74	88.7	7,868	150	164.7	27,125	375	389.7	151,850
30	44.7	1998	76	90.7	8,226	155	169.7	28,790	400	414.7	191,950
32	46.7	2180	78	92.7	8,593	160	174.7	30,500	425	439.7	193,300
34	48.7	2372	80	94.7	8,968	165	179.7	32,290	450	464.7	215,925
36	50.7	2570	82	96.7	9,351	170	184.7	34,100	475	489.7	239,790
38	52.7	2777	84	98.7	9,742	175	189.7	35,980	500	514.7	264,900
40	54.7	2992	86	100.7	10,140	180	194.7	37,095	550	564.7	318,900
42	56.7	3215	88	102.7	10,547	185	199.7	39,875	600	614.7	378,900
44	58.7	3446	90	104.7	10,962	190	204.7	41,900	650	684.7	441,800
46	60.7	3684	92	106.7	11,385	195	209.7	43,970	700	714.7	510,800
48	62.7	3931	94	108.7	11,816	200	214.7	46,090	750	764.7	584,800
50	64.7	4186	96	110.7	12,254	210	224.7	50,490	800	814.7	663,750
52	66.7	4449	98	112.7	12,701	220	234.7	55,060	900	914.7	836,700
54	68.7	4720	100	114.7	13,156	230	244.7	59,860	1000	1014.7	1,029,650

Another somewhat similar formula to D'Arcy's was published by Mr. J. E. Johnson, Jr., in the July 27, 1899, issue of the American Machinist, and which is

$$P_1^2 - P_2^2 = \frac{0.0006\, q^2 l}{d^5} \tag{97}$$

where,

P_1 = absolute initial air pressure in pounds;
P_2 = absolute terminal air pressure in pounds;
q = free air equivalent in cubic feet per minute of volume passing through the pipe;
l = length of pipe in feet;
d = diameter of pipe in inches.

Tables 24* and 25* are given to facilitate the use of Mr. Johnson's formula, and, in order to make clear the use of these tables, the following typical problems are solved.

Example 1. — Suppose 2000 cubic feet of free air under 90 pounds is required 1000 feet from the receiver. What size piping should be installed that will meet the requirements with 100 pounds initial pressure?

Substituting in formula:

$$(114.7)^2 - (104.7)^2 = \frac{0.0006 \times 2000^2 \times 1200}{d^5}$$

$$d^5 = \frac{0.0006 \times 2000^2 \times 1200}{114.7^2 - 104.7^2}$$

From Table 25,

$$114.7^2 = 13156$$
$$104.7^2 = 10962$$

Hence, $P_1^2 - P_1^2 = 2194$ for 1000 feet of pipe, or 219.4 for 100 feet.

Referring to Table 24, $P_1^2 - P_2^2$ for 1000 cubic feet of free air per minute through 3-inch pipe is equal to 247 and for $3\frac{1}{2}$-inch pipe is equal to 114. Therefore, $3\frac{1}{2}$-inch pipe should be used.

* Laidlaw-Dunn-Gordon Co.'s Catalogue.

Example 2. — Suppose 2000 feet of $3\frac{1}{2}$-inch pipe is already installed, and the equivalent of 1000 cubic feet of free air per minute under 70 pounds pressure is desired at the end. What receiver pressure is necessary?

Substituting in formula:

$$P_1^2 - 84.7^2 = \frac{0.0006 \times 1000^2 \times 2000}{3.5^2}$$

$$P_1^2 = \frac{0.0006 \times 1000^2 \times 2000}{3.5^2} + 84.7^2$$

Referring to Table 24,

$P_1^2 - P_2^2 = 455$ for 100 feet of pipe, or 9100 for 2000 feet.

From Table 25,

$$P_2^2 = (84.7)^2 = 7174$$

Now

$$P_1^2 = P_2^2 + (P_1^2 - P_2^2) = 16274.$$

In Table 25, 16,274 is between the squares of 110 and 115 pounds gauge pressures. Therefore, an initial pressure of approximately 113 pounds will be necessary.

Example 3. — Suppose an air compressor having a free air per minute capacity of 500 cubic feet is discharging against a pressure of 100 pounds into a $2\frac{1}{2}$-inch pipe line 1000 feet long, what will be the terminal pressure?

Substituting in formula:

$$114.7^2 - P_2^2 = \frac{0.0006 \times 500^2 \times 1000}{2.5^2}$$

$$P_2^2 = 114 - \frac{0.0006 \times 500^2 \times 1000}{2.5^2}$$

From Table 25,

$$P_1^2 = 114.7^2 = 13156.$$

From Table 24,

$$P_1^2 - P_2^2 = 154 \text{ for 100 feet of pipe,}$$

or 1540 for 1000 feet.

Now

$$P_2^2 = P_1^2 - (P_1^2 - P_2^2)$$
$$= 11,616.$$

In Table 25, 11,616 is between the squares of 92 and 94 pounds gauge pressures. Therefore, the terminal pressure would be about 93.5 pounds.

Loss Due to Valves, Tees and Elbows. — Thus far in the discussion, we have assumed clean, straight pipe, free from valves, tees and elbows. All of these fittings, when installed in the pipe line, create additional friction loss and, consequently, their use should be dispensed with whenever possible. There are practically no experimental data to be had regarding the amount of loss caused by the addition of these fittings, with the exception of one or two tables given in air compressor manufacturers' catalogues.

The Ingersoll-Rand Co., in their catalogue No. 74, state that the reduction of pressure caused by globe valves is equivalent to that caused by the following additional lengths of straight pipe:

Diameter of pipe	1	1½	2	2½	3	3½	4	5	6	7	8	10	12	15	18	20	22	24
Additional length	2	4	7	10	13	16	20	28	36	44	53	70	88	115	143	162	181	200

and the reduction of pressure caused by either elbows or tees is equal to two-thirds of that caused by globe valves, or,

Diameter of pipe	1	1½	2	2½	3	3½	7	8	10	12	15	18	20	22	24
Additional length	2	3	5	7	9	11	30	35	47	59	77	96	108	120	134

The more abrupt the change in direction of the pipe line, the greater will be the retarding effect upon the contained air and, consequently, the greater will be the loss. The resistance caused by an elbow increases as its radius of curvature decreases; therefore, long sweep elbows or bent pipe should always be chosen. The following (Table 26), taken from the catalogue of the Norwalk Iron Works Co., shows a relation of additional length of pipe to elbow radius in terms of pipe diameter.

Table 27 shows the standard dimensions and weights of

wrought-iron pipe, all of which will be found useful in preparing design of air lines.

In Tables 27 and 28 are specifications of screw and couplings for wrought pipe.

TABLE 26

Radius of elbow in terms of diameter of pipe	5	3	2	1½	1¼	1	¾	½
Equivalent length of straight pipe in terms of its diameter	7.85	8.24	9.03	10.36	12.72	17.51	35.09	121.2

TABLE 27

TABLE OF STANDARD DIMENSIONS OF WROUGHT-IRON PIPE

Nominal inside diameter	Actual inside diameter	Actual outside diameter	Internal area, square inches	External area, square inches	U. S. gallon per foot of pipe	Weight of pipe per lineal foot
Inches	Inches	Inches	Sq. ins.	Sq. ins.	Gallons	Pounds
⅛	0.270	0.405	0.057	0.1288	0.0029	0.24
¼	0.364	0.540	0.104	0.2290	0.0054	0.42
⅜	0.493	0.675	0.191	0.3578	0.0099	0.56
½	0.622	0.840	0.304	0.554	0.0158	0.84
¾	0.824	1.050	0.533	0.866	0.0277	1.12
1	1.048	1.315	0.861	1.358	0.0447	1.67
1¼	1.380	1.660	1.496	2.164	0.0777	2.24
1½	1.610	1.900	2.036	2.835	0.1058	2.68
2	2.067	2.375	3.356	4.430	0.1743	3.61
2½	2.468	2.875	4.780	6.492	0.2483	5.74
3	3.067	3.500	7.383	9.621	0.3835	7.54
3½	3.548	4.000	9.887	12.566	0.5136	9.00
4	4.026	4.500	12.730	15.904	0.6613	10.66
4½	4.508	5.000	15.961	19.635	0.829	12.34
5	5.045	5.563	19.986	24.301	1.038	14.50
6	6.065	6.625	28.890	34.472	1.500	18.76
7	7.023	7.625	38.738	45.664	2.012	23.27
8	7.981	8.625	50.027	58.426	2.599	28.18
9	8.927	9.635	62.730	72.760	3.259	33.70
10	10.018	10.75	78.823	90.763	4.095	40.06
11	11.000	11.75	95.033	108.434	4.937	45.02
12	12.000	12.75	113.098	127.677	5.875	49.00
13	13.25	14	137.887	153.938	7.163	54.00
14	14.25	15	159.485	176.715	8.285	58.00
15	15.25	16	182.665	201.062	9.489	62.00

Air Line Design. — There are other factors besides pure efficiency of air transmission that should be considered in a pipe line design. To reduce to the absurd, it would be poor economy to transmit 250 cubic feet of free air per minute through a 10-

TABLE 28

STANDARD DIMENSIONS OF COUPLINGS FOR STEAM, GAS AND WATER PIPE — BLACK AND GALVANIZED

Size of pipe, nominal, inside diameter	Inside diameter of coupling	Outside diameter of coupling	Outside area of coupling	Length of coupling	Threads per inch of screw	Average weight of coupling in pounds
Inches	Inches	Inches	Sq. ins.	Inches		
$\frac{1}{8}$	$\frac{11}{32}$	$\frac{19}{32}$	0.276	$\frac{13}{16}$	27	0.031
$\frac{1}{4}$	$\frac{15}{32}$	$\frac{23}{32}$	0.405	$\frac{15}{16}$	18	0.046
$\frac{3}{8}$	$\frac{37}{64}$	$\frac{27}{32}$	0.559	$1\frac{1}{16}$	18	0.078
$\frac{1}{2}$	$\frac{23}{32}$	1	0.785	$1\frac{5}{16}$	14	0.124
$\frac{3}{4}$	$\frac{59}{64}$	$1\frac{21}{64}$	1.382	$1\frac{9}{16}$	14	0.250
1	$1\frac{11}{64}$	$1\frac{9}{16}$	1.917	$1\frac{13}{16}$	$11\frac{1}{2}$	0.455
$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{61}{64}$	2.953	$2\frac{1}{8}$	$11\frac{1}{2}$	0.562
$1\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{7}{32}$	3 832	$2\frac{3}{8}$	$11\frac{1}{2}$	0.800
2	$2\frac{7}{32}$	$2\frac{3}{4}$	5.939	$2\frac{5}{8}$	$11\frac{1}{2}$	1.250
$2\frac{1}{2}$	$2\frac{21}{32}$	$3\frac{9}{32}$	8.419	$2\frac{7}{8}$	8	1.757
3	$3\frac{1}{4}$	$3\frac{15}{16}$	12.177	$3\frac{1}{8}$	8	2.625
$3\frac{1}{2}$	$3\frac{25}{32}$	$4\frac{7}{16}$	15.466	$3\frac{5}{8}$	8	4.000
4	$4\frac{17}{64}$	5	19 635	$3\frac{5}{8}$	8	4.125
$4\frac{1}{2}$	$4\frac{3}{4}$	$5\frac{1}{2}$	23.758	$3\frac{5}{8}$	8	4.875
5	$5\frac{9}{32}$	$6\frac{7}{32}$	30.347	$4\frac{1}{8}$	8	8.437
6	$6\frac{11}{32}$	$7\frac{5}{16}$	41.991	$4\frac{1}{8}$	8	10.625
7	$7\frac{3}{8}$	$8\frac{5}{16}$	54.255	$4\frac{1}{8}$	8	11.270
8	$8\frac{3}{8}$	$9\frac{5}{16}$	68.078	$4\frac{5}{8}$	8	15.150
9	$9\frac{7}{16}$	$10\frac{3}{8}$	84.541	$5\frac{1}{8}$	8	17.820
10	$10\frac{7}{16}$	$11\frac{21}{32}$	106.688	$6\frac{1}{8}$	8	27.700
11	$11\frac{15}{32}$	$12\frac{21}{32}$	125.78	$6\frac{1}{8}$	8	33.250
12	$12\frac{7}{16}$	$13\frac{7}{8}$	151.20	$6\frac{1}{8}$	8	43.187
13	$13\frac{11}{16}$	$15\frac{1}{16}$	178.16	$6\frac{1}{8}$	8	49.280
14	$14\frac{23}{32}$	$16\frac{3}{8}$	210.60	$6\frac{1}{8}$	8	63.270
15	$15\frac{11}{16}$	$17\frac{3}{8}$	237.10	$6\frac{1}{8}$	8	66.000

inch line, simply because the losses would be negligible. The interest on the investment, depreciation, and up-keep are really part and parcel of operating economy, but, unfortunately, are entirely lost sight of and sacrificed by "efficiency" enthusiasts in their designs of power plants, pipe lines, etc.

To illustrate the proper method of procedure in long air-line design, consider the following hypothetical case:

Suppose we are required to deliver the equivalent of 5000 cubic feet of free air per minute under 80 pounds pressure at the end of 2000 feet of pipe, with 100 pounds maximum on the receiver. Assume further, that the compressor installed or contemplated is the cross compound Corliss condensing steam end, two-stage air end and has a steam consumption of 15 pounds per I.H.P. hour. The boiler evaporation is, say, 10 pounds of steam per pound of coal, and the coal is worth $3.00 per ton delivered at the furnace. The conditions require four 90-degree bends and two globe valves. What size of piping is best suited to the requirements?

We are limited to a pressure drop of 20 pounds gauge or 34.7 pounds absolute; consequently, we must install a pipe line whose losses do not exceed this amount. First, then, determine accurately the size pipe having this maximum loss by substituting Mr. Johnson's formula:

$$114.7^2 - 94.7^2 = \frac{0.0006 \times 5000^2 \times 2000}{d^5}$$

Solving,

$$d^5 = 7163$$
$$d = 5.9$$

No pipe is manufactured of the above inside diameter; therefore the smallest commercial line possible, the limitations considered, is pipe of 6.065 inch inside diameter, or what is known as 6-inch standard pipe. In order to find the actual losses entailed in the use of this size pipe, again substitute in the formula as follows:

$$P_1^2 - 94.7^2 = \frac{0.0006 \times 5000^2 \times 2000}{6.065^5}$$

Solving

$P_1^2 = 12659$

$P_1 = 112.6$ or a gauge pressure of nearly 98 pounds per square inch.

Added to this, the loss caused by the four elbows and two globe valves makes initial pressure amount to 99.4 pounds.

The Indicated Horse Power required to compress 5000 cubic feet of free air per minute up to 99.4 pounds pressure is (Table 17) $5000 \times 0.176 = 880$ and which necessitates the generation of $880 \times 15 = 13{,}200$ pounds of steam per hour at the expense of 1320 pounds of coal. The yearly fuel cost for ten hours a day operation (Sundays excluded) then is

$$\frac{1320 \times 10 \times 313 \times \$3}{2000} = \$6197.40$$

The cost of a six-inch pipe line and fittings, not including excavation or labor, would be about $1000.00. Taking interest and depreciation into consideration, the total operating expenses tabulate as follows:

Fuel cost for one year	$6197.40
Six per cent interest on pipe line	60.00
Ten per cent depreciation on pipe line	100.00
	$6357.40

The cost now of a 7-inch pipe line and fittings, likewise exclusive of labor and excavation which would amount to practically the same as for 6-inch pipe, is about $1400.00. By the same method of calculating as previously employed, we find that for this size pipe line the operating, interest and depreciation charges tabulate as follows:

Fuel cost for one year	$6076.00*
Six per cent interest on pipe line	84.00
Six per cent depreciation on pipe line	140.00
	$6300.00

* Assumed 15.5 pounds steam consumption since the compressor is only 90 per cent loaded.

inch line, simply because the losses would be negligible. The interest on the investment, depreciation, and up-keep are really part and parcel of operating economy, but, unfortunately, are entirely lost sight of and sacrificed by "efficiency" enthusiasts in their designs of power plants, pipe lines, etc.

To illustrate the proper method of procedure in long air-line design, consider the following hypothetical case:

Suppose we are required to deliver the equivalent of 5000 cubic feet of free air per minute under 80 pounds pressure at the end of 2000 feet of pipe, with 100 pounds maximum on the receiver. Assume further, that the compressor installed or contemplated is the cross compound Corliss condensing steam end, two-stage air end and has a steam consumption of 15 pounds per I.H.P. hour. The boiler evaporation is, say, 10 pounds of steam per pound of coal, and the coal is worth $3.00 per ton delivered at the furnace. The conditions require four 90-degree bends and two globe valves. What size of piping is best suited to the requirements?

We are limited to a pressure drop of 20 pounds gauge or 34.7 pounds absolute; consequently, we must install a pipe line whose losses do not exceed this amount. First, then, determine accurately the size pipe having this maximum loss by substituting Mr. Johnson's formula:

$$114.7^2 - 94.7^2 = \frac{0.0006 \times 5000^2 \times 2000}{d^5}$$

Solving,

$$d^5 = 7163$$
$$d = 5.9$$

No pipe is manufactured of the above inside diameter; therefore the smallest commercial line possible, the limitations considered, is pipe of 6.065 inch inside diameter, or what is known as 6-inch standard pipe. In order to find the actual losses entailed in the use of this size pipe, again substitute in the formula as follows:

$$P_1^2 - 94.7^2 = \frac{0.0006 \times 5000^2 \times 2000}{6.065^5}$$

Pipe lines are constructed of cast iron, wrought iron or steel riveted pipe, and connecting joints are made with either sleeve or flange couplings. The typical air line is wrought pipe with sleeve couplings. Bends and fittings should be installed only where absolutely necessary for reasons as before given. Provision should be made by blowing out water at the end of the line even if an additional valve and fittings are necessary.

CHAPTER XIII

FLOW OF WATER IN PIPES

Theoretically the flow of water through pipes is in accordance with the fundamental formula:

$$v = \sqrt{2\,gh} \tag{98}$$

where

v = velocity of flow in feet per second;
g = acceleration in feet per second due to gravity;
h = head in feet at the pipe end causing the flow;

Therefore, solving (98) for h we have

$$h = \frac{v^2}{2\,g} \tag{99}$$

If there were no friction losses or no entrance losses this formula would tell the whole story of the flow of water; but, like air, the flow of water through a pipe line is accompanied by a loss of head or pressure proportional to the length and diameter of pipe, quantity of water, condition of the inner surface of the pipe, etc. Many experiments have been performed with a view of establishing constants and formulas, notably those of D'Arcy, but errors in calculations of anywhere from 5 to 15 per cent are unavoidable.

The laws governing the flow of water are summed up as follows:

1. The loss in head due to friction is directly proportional to the length of pipe through which the water flows.

2. The friction loss increases with the decrease in pipe diameter.

3. The loss increases nearly as the square of the velocity of flow.

4. The loss is independent of the pressure of the water.

5. The loss increases with the roughness of the interior surface of the pipe.

All of these laws are expressed in the well-known formula:

$$H_1 = f \frac{l}{d} \frac{v^2}{2g} \tag{100}$$

where

H_1 = loss head in feet due to friction;
f = friction factor which varies with the diameter and nature of the inner surface of pipes;
l = length of pipe in feet;
d = diameter of pipe in feet;
$\frac{v^2}{2g}$ = velocity head.

The factor f is the uncontrollable quantity in the formula. It varies not only with the condition of the inner surface of the pipe, but also with the pipe diameter and the velocity of flow of water in the pipe. In Table 29 are given experimental values of f compiled from discussions of various authorities. The probable error in the values tabulated amount to about ten per cent. For approximate calculations the mean value of f may be taken as 0.02.

TABLE 29

Pipe diameter in feet	Velocity in feet per second						
	1	2	3	4	6	10	15
0.05	0.047	0.041	0.037	0.034	0.031	0.029	0.028
0.1	0.038	0.032	0.030	0.028	0.026	0.024	0.023
0.25	0.032	0.028	0.026	0.025	0.024	0.022	0.021
0.5	0.028	0.026	0.025	0.023	0.022	0.020	0.019
0.75	0.026	0.025	0.024	0.022	0.021	0.019	0.018
1.0	0.025	0.024	0.023	0.022	0.020	0.018	0.017
1.25	0.024	0.023	0.022	0.021	0.019	0.017	0.016
1.5	0.023	0.022	0.021	0.020	0.018	0.016	0.015
1.75	0.022	0.021	0.020	0.018	0.017	0.015	0.014
2.0	0.021	0.020	0.019	0.017	0.016	0.014	0.013
2.5	0.020	0.019	0.018	0.016	0.015	0.013	0.012
3.0	0.019	0.018	0.016	0.015	0.014	0.013	0.012
3.5	0.018	0.017	0.016	0.014	0.013	0.012	......
4.0	0.017	0.016	0.015	0.013	0.012	0.011	......
5.0	0.016	0.015	0.014	0.013	0.012	......	......
6.0	0.015	0.014	0.013	0.012	0.011	......	......

In Table 30 is given the capacity in gallons of water per minute discharged at various velocities, also the friction head in feet encountered. The values given are for lengths of 100 feet. The friction loss in pounds may be computed by multiplying the tabular values by 0.434.

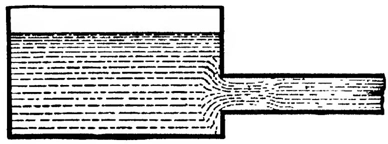


FIG. 114.

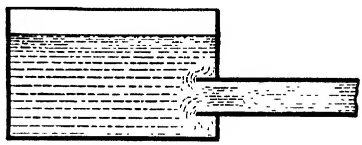


FIG. 115.

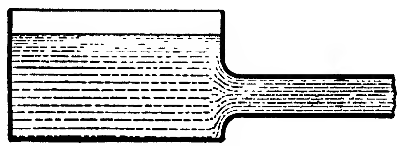


FIG. 116

To determine the total loss occurring in straight pipe of uniform diameter, the loss of entrance must be added to the friction. The entrance loss depends upon the shape of the pipe end. The straight standard end, the inward projecting end, and the bell end are shown in Figs. 114, 115 and 116, respectively. The loss is greatest in the second named and least in the last named. The entrance loss may be expressed by the formula: —

$$H_2 = C \frac{v^2}{2g} \qquad (101)$$

where

C = constant, the value of which depends upon the pipe and design;

$\frac{v^2}{2g}$ = velocity head as before.

The values of C for the end shown in Fig. 114 is 0.93, for that shown in Fig. 114 is 0.49, and for Fig. 116 is 0. It is customary to use 0.5 as the value of C in approximate calculations or where

TABLE 30

CAPACITY IN GALLONS PER MINUTE DISCHARGED AT VELOCITIES IN FEET PER SECOND, FROM 3 TO 15. ALSO FRICTION HEAD IN FEET PER 100 FEET LENGTH OF PIPE

Diam. pipe	1-Inch		2-Inch		3-Inch		4-Inch		5-Inch		6-Inch	
Velocity	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction
3	7 34	4 08	29 37	2.04	66 09	1.36	117 50	1.02	183 63	0.816	264.24	0 68
4	9.79	6 83	39 16	3 41	88 12	2.27	156 67	1 71	244.84	1.36	352.32	1.13
5	12 24	10 2	48 95	5 12	110.15	3 41	195.70	2 56	306 05	2.05	440.40	1.70
6	14 68	14.3	58 74	7.16	132 18	4 78	235.84	3 58	367.26	2 86	528.48	2 38
7	17.13	19 0	68 53	9 54	154 21	6.36	274.98	4 77	428 47	3 81	616.56	3.18
8	19 58	24 5	78 32	12 2	176 24	8 16	314 12	6 12	489.68	4.90	705.64	4.08
8½	20.80	27 4	83 23	13 7	187 25	9.15	333.75	6 86	520.61	5 49	749.01	4.57
9	22 03	30 5	88 11	15 2	198 27	10 1	352.26	7 64	550 89	6 11	793 72	5 09
9½	23 25	33 8	93 00	16 9	209 24	11 2	371 90	8.46	581.25	6 77	837.08	5.61
10	24 48	37 3	97 90	18 6	220 30	12 4	391 40	9 33	612 10	7 46	881.80	6 21
10½	25 70	40 9	102 80	20 4	231 31	13 6	411 05	10.2	642.43	8 19	925 20	6.82
11	26 92	44 7	107 69	22 3	242 33	14 9	430 54	11 1	673 31	8 95	969.88	7 45
11½	28 15	48 7	112 58	24 3	253 34	16 2	450 20	12 1	703.62	9 74	1013.3	8.11
12	29 37	52 8	117.48	26 4	264 36	17 6	470 68	13 2	734.52	10 5	1057 9	8 80
13	31.82	61 5	127 27	30 7	286 39	20 5	509.82	15 3	795 73	12.3	1145 0	10 2
14	34 27	71 0	137 06	35 5	308 42	23 7	548 96	17 7	856 94	14.2	1233 1	11 8
15	36 72	81.0	146 85	40 5	330 45	27 0	587.10	20.3	918.15	16.2	1321 2	13.5

Diam. pipe	7-Inch		8-Inch		9-Inch		10-Inch		12-Inch		14-Inch	
Velocity	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction
3	359.79	0 583	470.04	0 510	594.78	0 453	734 40	0.408	1057.5	0.347	1439.0	0.291
4	479.72	0 976	626 72	0 854	793 04	0 759	979.20	0 683	1410 0	0 581	1919.7	0.488
5	599 65	1 46	783.40	1 28	991.30	1.13	1224.0	1.02	1762.6	0.871	2399 4	0.731
6	719 58	2 05	940 08	1 79	1189 5	1.59	1468 8	1.43	2115 1	1.21	2878.0	1.02
7	839.51	2.72	1096.7	2 38	1388 8	2.12	1713.6	1 90	2467 6	1.62	3358 7	1.36
8	959.44	3 49	1253 4	3 06	1586.0	2.72	1958.4	2 45	2820 1	2.08	3838.4	1.75
8½	1019.4	3 92	1331 5	3.43	1685.0	3.05	2080 8	2 74	2996.3	2.33	4078.3	1.96
9	1079.4	4.36	1410.1	3.82	1784.3	3.40	2203 2	3.05	3172.7	2.60	4318.1	2.18
9½	1139 4	4.83	1488 0	4.23	1883.5	3.76	2325.6	3.38	3348.9	2.88	4558.0	2.42
10	1199.3	5.33	1566.8	4.66	1982 6	4.14	2448.0	3.73	3525.2	3.17	4798.0	2.66
10½	1259:3	5.84	1645.8	5.22	2082.7	4.55	2570.8	4.09	3701.4	3 48	5037.7	2.92
11	1319.2	6 39	1723.5	5 59	2181 9	4.97	2692.8	4.47	3877.7	3.80	5277.5	3.19
11½	1379.2	6.95	1801.5	6.08	2280.0	5.41	2815.2	4.87	4053.8	4.14	5517.4	3.48
12	1439.2	7.54	1880 2	6 60	2379.1	5 87	2937.6	5.28	4230.2	4.49	5757.2	3.77
13	1559.1	8.79	2036.8	7.00	2577.4	6.84	3182.4	6.15	4582.8	5.23	6237.8	4.40
14	1679.0	10.1	2193.5	8.87	2776.6	7.88	3427.2	7.10	4935.4	6.03	6717.5	5.06
15	1799.0	11.6	2350.2	10.1	2974.9	9.00	3672.0	8.10	5287.8	6.89	7107.2	5.79

TABLE 30 — (Continued)

Capacity in Gallons per Minute Discharged at Velocities in Feet per Second, from 3 to 15. Also Friction Head in Feet per 100 Feet Length of Pipe

Diam. pipe	15-Inch		18-Inch		20-Inch		22-Inch		24-Inch		26-Inch	
Velocity	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction
3	1652 2	0 272	2,379 7	0 227	2,937 0	0 204	3,554 1	0 185	4,230 3	0 170	4,964.2	0.157
4	2203.0	0 455	3,172 6	0 379	3,916 0	0.342	4,739 8	0 310	5,640 0	0.284	6,619.0	0.262
5	2754 7	0 682	3,965.5	0 569	4,896 0	0 512	5,924.5	0.465	7,050.8	0 426	8,274.7	0.394
6	3304.4	0 955	4,758 4	0.795	5,875 0	0 717	7,108 2	0.651	8,460 6	0 597	9,929.5	0.550
7	3855 2	1 27	5,552 3	1.06	6,854 0	0 954	8,293 9	0 866	9,870 3	0.794	11,583	0.753
8	4406 9	1 63	6,345.2	1 36	7,833 0	1.22	9,478 6	1 11	11,280	1 01	13,238	0.940
8½	4688 1	1 82	6,741 9	1 52	8,323 6	1 37	10,071	1.25	11,985	1.14	14,066	1.05
9	4957.7	2 04	7,138 1	1 70	8,812 0	1 53	10,663	1 39	12,690	1 27	14,893	1.17
9½	5232 1	2.25	7,534 8	1 83	9,302 6	1 69	11,255	1 54	13,395	1 40	15,721	1.30
10	5508.4	2.50	7,931 0	2.07	9,792	1.87	11,848	1 69	14,100	1 55	16,548	1.43
10½	5783.4	2 73	8,328 8	2 27	10,281	2.05	12,440	1 86	14,805	1.70	17,375	1.57
11	6058.2	2 98	8,724 9	2 48	10,771	2.24	13,033	2 03	15,510	1 86	18,202	1.72
11½	6334 6	3 25	9,121 7	2 70	11,258	2.43	13,625	2 21	16,215	2 03	19,029	1.87
12	6609 9	3 52	9,517 8	2.93	11,750	2.64	14,217	2 40	16,920	2.20	19,857	2.03
13	7160.6	4 10	10,310	3.42	12,729	3.08	15,402	2 79	18,330	2.56	21,511	2.36
14	7711.4	4 73	11,104	3 93	13,708	3.55	16,587	3 22	19,740	2 95	23,166	2.73
15	8262 1	5 40	11,897	4 50	14,688	4.05	17,772	3 68	21,150	3 37	24,824	3.11

Diam. pipe	28-Inch		30-Inch		32-Inch		36-Inch		42-Inch		48-Inch	
Velocity	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction	Capacity	Friction
3	5,757.2	0 146	6,609	0.136	7,519.7	0.127	9,518	0 113	12,954	0.097	16,921	0.085
4	7,676.2	0 244	8,812	0 227	10,026	0.213	12,690	0 189	17,272	0.163	22,561	0.143
5	9,596.3	0.366	11,015	0.341	12,532	0 320	15,863	0 284	21,590	0 244	28,201	0 213
6	11,514	0.512	13,218	0 478	15,039	0 447	19,033	0 397	25,908	0.341	33,841	0 298
7	13,434	0.681	15,421	0 636	17,546	0.591	22,208	0 528	30,226	0.454	39,482	0 397
8	15,353	0.875	17,624	0.816	20,052	0 764	25,381	0 679	34,544	0.583	45,122	0.510
8½	16,316	0 980	18,725	0 915	21,306	0.857	26,967	0 760	36,704	0.653	47,942	0.571
9	17,273	1 09	19,827	1.01	22,559	0 954	28,554	0 847	38,863	0.728	50,762	0 636
9½	18,231	1.21	20,928	1.12	23,812	1.06	30,140	0.938	41,022	0.806	53,582	0.694
10	19,192	1 33	22,030	1 24	25,065	1.16	31,726	1.03	43,181	0 888	56,403	0.778
10½	20,150	1.46	23,131	1.36	26,319	1.28	33,313	1.13	45,340	0.975	59,223	0.851
11	21,111	1 60	24,233	1.49	27,572	1.40	34,899	1.16	47,499	1.06	62,043	0.930
11½	22,069	1.74	25,338	1 62	28,825	1 52	36,485	1.35	49,658	1.16	64,863	1.00
12	23,030	1.89	26,436	1.76	30,079	1.65	38,072	1.46	51,817	1 26	67,683	1.10
13	24,950	2.20	28,639	2.05	32,585	1.92	41,244	1.70	56,135	1.46	73,324	1.28
14	26,869	2.53	30,842	2.37	35,092	2.21	44,417	1.97	60,453	1.69	78,964	1.48
15	28,788	2.89	33,045	2.70	37,598	2.53	47,590	2.24	64,771	1.93	84,604	1.69

the shape of the end is not stated. The entrance loss is very small in proportion to friction loss in very long pipe lines, but in short lengths, the entrance loss is often the greater. Therefore, to obtain the total head necessary to force the water through the pipe, all losses must be added to the velocity head or

$$H = h + H_1 + H_2 \tag{102}$$

Substituting the values previously determined, we have

$$H = \frac{v^2}{2g} + f\frac{l}{d}\frac{v^2}{2g} + C\frac{v^2}{2g}$$

simplifying,

$$H = \frac{v^2}{2g}\left(1 + f\frac{l}{d} + C\right) \tag{103}$$

The above equation (100) is the fundamental formula for the flow of water in clean, straight pipe of uniform diameter having close joints.

Loss in Bends and Elbows. — Like air, whenever the direction of flow of water is changed there result additional losses to be overcome. Take an elbow, for instance: the water traveling in a straight line impinges on the outer wall of the bend, increasing the pressure along that surface and in a direction away from the center of the radius of curvature. Eddying motions, with the resulting impact of water particles occur, and energy is transformed into heat which is dissipated. The loss in long, easy bends is small, and is considerably greater in proportion for elbows in small pipes than for large ones.

The laws stated at the beginning of the chapter for losses by friction in straight pipe, apply also for curvature losses. Therefore, we may write the following formula:

$$H_3 = f_1\frac{l_1}{d}\frac{v^2}{2g} \tag{104}$$

where

H_3 = loss in feet;

f_1 = curve factor, the value of which depends upon the ratio of radius of curve R, to diameter of pipe d;

d = diameter of pipe in feet;

$\frac{v^2}{2g}$ = velocity head.

We know very little about the value of f_1 and what we do know is obtained from experiments performed either on bends in hose or curves without joints. In actual practice, poorly made joints must be contended with, and it is quite probable that losses from this source are much greater than purely curvature losses.

Professor Merriman in his *Treatise on Hydraulics* has computed from Weisbach's formula of 90-degree curve losses, the following values of f_1 for various ratios of curve radii to pipe diameters:

For R/d =	20	10	5	3	2	1.5	1
f_1 =	0.004	0.008	0.016	0.03	0.047	0.072	0.184

Weisbach's formula is accurately applicable only to bends in small pipes of smooth interior and free from joints. Professor Merriman has also computed the values from measurements made by Williams, Hubbell and Fenkell, on 12-inch and 30-inch cast-iron water mains in Detroit, Mich. For 30-inch pipe, the values are:

For R/d =	20	16	10	6	4	2.4
f_1 =	0.036	0.037	0.047	0.06	0.062	0.072

and for 12-inch pipe the values are:

For R/d =	4	3	2	1
f_1 =	0.05	0.06	0.06	0.2

These values are possibly more accurate than Weisbach's, because of the presence in the bends tested of the rougher surfaces and joints met with in practice.

In Table 31 are given the pressure losses in pounds per square inch in elbows or short bends. The table is based on Weisbach's formula, and conversion to feet head may be made by multiplying by 2.31.

TABLE 31

FRICTION OF WATER IN ELBOWS

(Pressure in Pounds per Square Inch to be Added for Each Elbow)

Gallons per min. delivered	Pipe sizes										
	2	2½	3	3½	4	5	6	7	8	9	10
5	0.002	...	...	...	.	...	....	....	....	.. .	....
10	0.006	0.003	...	...	...	...	....	....	....	....	....
15	0.014	0.005	...	...	...	...	....	....	....	....	....
20	0.025	0.012	0.005	...	. .	...	....	....	....	....	....
25	0.038	0.02	0.008	...	. .	..	.	....	....	. ..	....
30	0.055	0.028	0 011	...	...	...	....	....	....	....	....
35	0.076	0.037	0.015	0.009	...	...	....	....	...	....	....
40	0.098	0.049	0.02	0.011	0.007	...	....	....	....	...	....
45	0.125	0.062	0.026	0.015	0.009	. .	....	....	...	....	....
50	0.153	0.08	0.032	0.017	0 01	...	....	....	.. .	....	.. .
60	0.22	0 112	0.044	0 026	0.015	0.006	0.003	....	..	..	. ..
70	0.304	0.148	0.06	0.035	0.021	0.009	0.004	0.002	...	. .	....
75	0.35	0.172	0.072	0.04	0.024	0.01	0.005	0.003	....	..	. .
80	0.392	0.196	0.08	0.044	0.027	0.012	0.005	0.003	. .	.. .	...
90	0.50	0.248	0.104	0.06	0.035	0.014	0.007	0.004	....	....	...
100	0.612	0 32	0.128	0.068	0.043	0.017	0.008	0.005	0.003	0.002	....
125	0.970	0 48	0.20	0.112	0.067	0.027	0.013	0.007	0.004	0.003	0.002
150	1.39	0.685	0.286	0.16	0.096	0 039	0.019	0.01	0.006	0.004	0.003
175	1.90	0.935	0.390	0.218	0.132	0.053	0.026	0.014	0.009	0.005	0.004
200	2.44	1.128	0.512	0.272	0.172	0.068	0.032	0.02	0.011	0.007	0.005
250	3.86	1.91	0.80	0.446	0.268	0.109	0.025	0.029	0.017	0.011	0.007
300	5.56	2.74	1.14	0.64	0.384	0.156	0.076	0.042	0.025	0.016	0.01
350	.	3.77	1.58	0.88	0.530	0.215	0.103	0.057	0.034	0.022	0.014
400	. .	5.12	2.05	1.09	0.688	0.272	0.128	0.08	0.044	0.028	0.018
450	. .	6.20	2.58	1.45	0.870	0.352	0.170	0.094	0.057	0.036	0.023
500	. .	7.64	3.20	1.78	1.07	0.436	0.208	0.116	0.068	0.044	0.028
750	...	. .	...	...	2.42	0.970	0.470	0.260	0.156	0.10	0.063
1000	. .	.	...	...	4.28	1.74	0.832	0.464	0.272	0.176	0.112
1250	.	...	...	...	6.70	2.71	1.31	0.728	0.436	0.276	0.175
1500	. .	...	. .	. .	9.68	3.88	1.88	0.84	0.624	0.40	0.252

Loss in Valves. — The presence of globe valves, cocks or gate valves in a pipe line to regulate the flow of water causes additional losses. This from the fact that obstructions are offered to the flow, and the loss increases as the area of the opening is reduced by closing the valve. Of the three types of valves mentioned, the gate type is the least harmful, and, consequently, should be used wherever efficiency of flow is required. Thus the throttling loss is expressed by the formula:

$$H_4 = C_1 \frac{v^2}{2g} \qquad (105)$$

where

H_4 = head loss in feet;

C_1 = constant the value of which depends upon the area open to flow,

$\frac{v^2}{2g}$ = velocity head. The velocity v is that in the pipe and not that through the opening in the valve.

From Weisbach's experiments, the following valves of C have been computed. If a is the distance the gate valve is closed, then,

For a/d =	0	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
C_1 =	0	0.07	0.26	0.81	2.1	5.5	17	98

Losses of Expansion or Contraction of Section. — Whenever the cross-sectional area in a water pipe is suddenly increased or decreased there occurs a loss of head due to formation of eddies. Sudden expansion or contraction of section may be regarded in the same manner as a partially closed gate valve in the line. The loss due to these causes is expressed by the formula:

$$H_5 = C_2 \frac{v^2}{2g} \tag{106}$$

where

H_5 = loss in feet;

$C_2 = \frac{\text{area of pipe}}{\text{area of contracted section}}$ for sudden expansion or equals $\frac{\text{area of contracted section}}{\text{area of pipe}}$ for sudden contraction;

$\frac{v^2}{2g}$ = velocity head. For sudden contraction of area, v = velocity in the smaller area while for sudden expansion, v = velocity in the larger section.

It is customary to use formula (106) as denoting the sum of the valve loss and sudden contraction losses.

Velocity. — Thus it is seen that, in making calculations of losses or computing pipe sizes, it is necessary that the velocity (v) of water travel be known. In Formula (102), we have considered the velocity head, the friction and the entrance losses, but in actual practice the pipe line usually has elbows, valves, and often reductions in pipe diameter. To derive a formula that will embody all the essentials of practice, it is necessary that the losses calculated in Formulas (104), (105) and (106), be added to (102), making the total head as follows:

$$H = h + H_1 + H_2 + H_3 + H_4 \qquad (107)$$

Substituting the various equivalents, we have

$$H = \frac{v^2}{2g} + f\frac{l}{d}\frac{v_2}{2g} + C\frac{v^2}{2g} + f_1\frac{l_1}{d}\frac{v^2}{2g} + C_1\frac{v^2}{2g}$$

Simplifying:

$$H = \frac{v^2}{2g}\left(1 + f\frac{l}{d} + C + f_1\frac{l_1}{d} + C\right) \qquad (108)$$

solving for v we have

$$v = \sqrt{\frac{2gh}{1 + f\frac{l}{d} + C + f_1\frac{l_1}{d} + C_1}} \qquad (109)$$

For straight pipe without valves or elbows and with standard flush end entrance, the velocity is expressed by:

$$v = \sqrt{\frac{2gh}{1.5 + f\frac{l}{d}}} \qquad (110)$$

This is the formula usually employed in velocity computations. It is quite evident, however, that no direct calculations are possible because the friction factor f is necessarily a function of the velocity, v. To apply the formula to any specific case, a series of approximations or assumptions must be made until the value of f is in conformity with the calculated value of v. In other words, we first assume a value for f, substitute in the formula and determine the velocity, v; next consult the table and

ascertain the value of f corresponding to the just calculated v. Solve the formula with the new value of f substituted therein. This will give a new value for v, and the table is again consulted for a value of f corresponding to it. This operation is repeated until the tabular value of f is the same, or nearly, as that used in the formula.

Suppose, for instance, that we have a 6-inch pipe line 2000 feet long with a head of 10 feet, what is the mean velocity of discharge? Assume a value of 0.02 for f, and substitute various other values in formula (110) as follows,

$$v = \sqrt{\frac{64.32 \times 10}{1.5 + 0.02 \frac{2000}{0.5}}} = 2.79 \text{ feet per second.}$$

Referring to Table 29, the value of f for 6-inch pipe and 2.79 feet per second velocity is 0.025. Substituting this in the formula we have:

$$v = \sqrt{\frac{64.32 \times 10}{1.5 + 0.025 \frac{2000}{0.5}}}.$$

$$v = 2.52 \text{ feet per second.}$$

Again referring to the table, the value of f for 2.52 feet per second is 0.0255. Substitute this value, and we have:

$$v = \sqrt{\frac{64.32 \times 10}{1.5 + 0.0255 \frac{2000}{0.5}}}$$

$$v = 2.49+ \text{ feet per second.}$$

Referring to the table for the third time, the value of f is 0.0255, which is the value we have used. Therefore, 2.49 feet per second is the probable velocity.

Capacity. — The discharge capacity of a pipe in cubic feet per second can be found by substituting in the formula

$$q = av \tag{111}$$

where,

q = cubic feet of water per second;

a = cross sectional area of the pipe in square feet;

v = velocity in feet per second and is determined by method just previously described.

For a in Formula 111, we may substitute its equivalent, $\frac{1}{4}\pi d^2$, and for v its equivalent as written in 110. Expression 111 then becomes:

$$q = \frac{1}{4}\pi d^2 \sqrt{\frac{2gH}{1.5 + f\frac{l}{d}}} \tag{112}$$

This expression holds for straight pipe free from curves and valves. If these latter are installed in the line, the formula expressing quantity of discharge becomes:

$$q = \frac{1}{4}\pi d^2 \sqrt{\frac{2\,gH}{1.5 + f\frac{l}{d} + f_1\frac{l_1}{d} + C_1}} \tag{113}$$

The application of this formula is self-evident.

Pipe Diameter. — By transposing and solving 111 for d, as follows, we have

$$d^4 = \frac{8\,q^2}{\pi^2} \times \frac{1.5 + f\frac{l}{d}}{gH}$$

multiplying through by d

$$d^5 = \frac{8\,q^2}{\pi^2 g}(1.5d + fl)\frac{1}{H}$$

$$d = \sqrt{\frac{8}{\pi^2 g}(1.5d + fl)\frac{q^2}{H}} \tag{114}$$

which gives the diameter of the pipe in feet when the values of the other symbols are known.

The application of this formula is quite similar to that for velocity. The usual method of procedure is:

1. Assume $f = 0.02$ and substitute in the formula;

2. Neglect $(1.5\,d)$ in the right-hand member of the formula;

3. Solve the formula, ascertaining an approximate value of d;

4. Compute from $q = av$ the velocity corresponding to the approximate diameter;

5. Refer to Table 29 and ascertain the value of f corresponding to the above pipe diameter and velocity;

6. Substitute the new value of f and the approximate value of d in the right-hand member and solve.

Repeat this operation until component factors, that is, f and v, agree or nearly so, and the result will be a diameter size that will satisfy the conditions.

Design. — The remarks in the previous chapter pertaining to air-line design and construction apply also to water-line design and construction. Long water lines are usually made with cast-iron bell and spigot pipe laid beneath the ground surface to prevent freezing in cold weather.

CHAPTER XIV

A PROPERLY DESIGNED INSTALLATION

That a pumping plant designed in accordance with the principles given on the preceding pages will prove efficient in operation is forcibly demonstrated by the results obtained from the water works plant at Greenville, Miss. In order to illustrate the method of installation of the apparatus, a full and detailed description of the plant and other pertinent facts are given on the following pages.*

Prior to January 1, 1915, the municipally owned and operated water works plant at Greenville, Miss., was steam actuated as were the sewage disposal pumps. The equipment consisted, in brief, of a 900 cu. ft., cross-compound steam, single stage air compressor for pumping from three deep wells into surface storage reservoirs; one 3,000,000-gallon triple-expansion, direct acting steam pump for furnishing the water to the mains; and one 2000 gallon per minute vertical shaft centrigufal sewage pump located in a pit and operated by a tandem compound automatic engine with a quarter twist rope drive. There were also two 300 cu. ft. straight-line air compressors and two 750 gallon-per-minute simple direct-acting steam pumps reserved for emergencies. Steam was furnished by one 300 horse-power internally fired boiler and one 250 horse-power water-tube boiler.

An average of two million gallons of water and one million gallons of sewage were pumped each twenty four hours; the former against approximately 65 pounds pressure and the latter 45 pounds, the pressure varying somewhat with the stage of the water in the river into which the sewage was pumped. No tower and tank or elevated storage of any kind was pro-

* *An Electric Driven Water Works Plant* by E. M. Ivens in *Power*, Aug. 10, 1915.

vided, and this compelled continuous operation of the water pumps under widely varying conditions of load.

The fuel and labor costs amounted to nearly $20,000, annually and as this seemed excessive, it was decided by the city council to devise ways and means for an improvement. A proposition was received from the Delta Light & Traction Co., which owns the lighting franchise, offering to pump all water and sewage electrically, to bear the cost of the necessary machinery and its installation and to maintain the operating force, for approximately $14,000 per year. The price also included fuel and labor for banking the fires under the boilers as required by the State Board of Fire Underwriters. The proposition was promptly accepted and contract signed.

In the purchase of the electrically driven machinery by the Delta Light & Traction Co., price was obviously of secondary importance, economy of operation being the first consideration. After investigating various methods of pumping from deep wells and into mains and exercising the utmost care in the selection of types and sizes of apparatus, it was decided to install air lifts in the wells and utilize single-stage centrifugal pumps for supplying the mains. The equipment purchased consisted of two 549 cu. ft., duplex type air compressors operated through short belt drives by two 50 horse-power motors; two 1500-gallon-per-minute single-stage centrifugal pumps, each direct connected to a 75 horse-power motor; one 750 gallon-per-minute single-stage centrifugal pump direct connected to a 50 horse-power motor; one 2000 gallon-per-minute vertical shaft centrifugal sewage pump direct connected to a 75 horse-power motor; one 900 gallon-per-minute horizontal type centrifugal sewage pump direct connected to a 35 horse-power motor; and the necessary switchboard for control. All motors were of the 2300 volt, 3 phase, 60 cycle type.

Figure 117 is a floor plan of the plant and shows the location of all machinery, reservoirs, wells and piping. The dotted outlines indicate the old steam-driven equipment and the full lines, the new electrically driven machinery. Since the drawing

was made, a fourth well has been drilled near Reservoir No. 2, and connected to the receiver manifold in the same manner and with the same sized piping as are the other three wells.

Compressed Air Equipment. — Figure 118 is a cut illustrating the type of air compressors installed. Each machine has en-

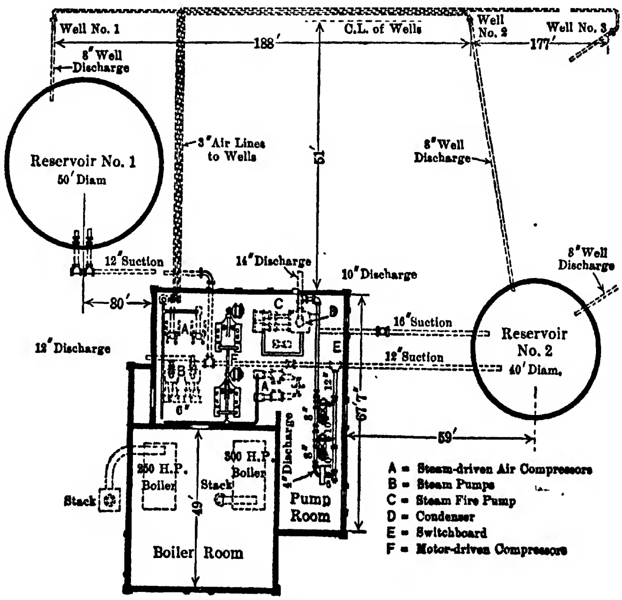


FIG. 117.

closed crank cases and all wearing surfaces except the inside of the air cylinders are lubricated by the "splash" system. Force feed lubricators serve the air pistons, cylinder walls, and valves.

The air discharge openings from each pair of air cylinders are connected, and valves so located that any compressor in the building (either steam or electric driven) may be made to furnish air to any one or to all the wells.

Air inlet pipes are brought to the outside of the building and the ends screened, thus insuring the drawing in of the coolest and cleanest air available.

Water Pumps. — The three centrifugal water pumps and the discharge piping are shown in Fig. 119.

The suction and discharge piping is such that water may be drawn from either reservoir and either one or all the pumps may be operated on the mains. A by-pass from the suction to the discharge is provided between the two larger pumps so that the discharge from one may be connected to the suction of the

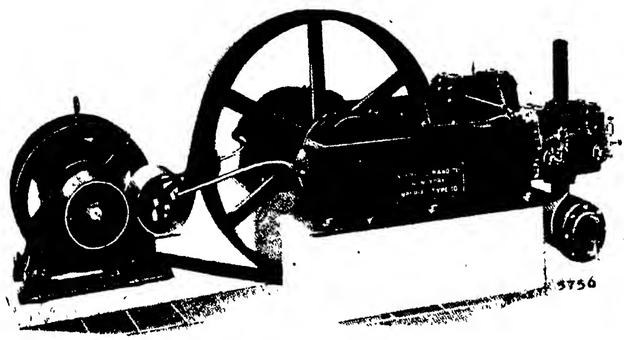


FIG. 118.

other, giving, in effect, one two-stage pump having the water capacity of one pump and the combined pressure capacity of both pumps. The object of this arrangement is to make available sufficient water pressure in the mains for ordinary fire-fighting purposes and thus obviate the necessity of starting the steam apparatus. In case of conflagrations, one or all of the steam pumps may be added, and banked fires are maintained under the boilers in readiness for just such emergencies.

A flow meter is installed in the discharge header and an integrating wattmeter is mounted on the switchboard, so that water-horse-power output and power input may both be readily calculated at any time and any falling off in efficiency can be quickly noted and the cause remedied if possible.

The Sewage Pumps. — Figure 120 is an interior view of the sewage pumping station, showing the two pumping units with

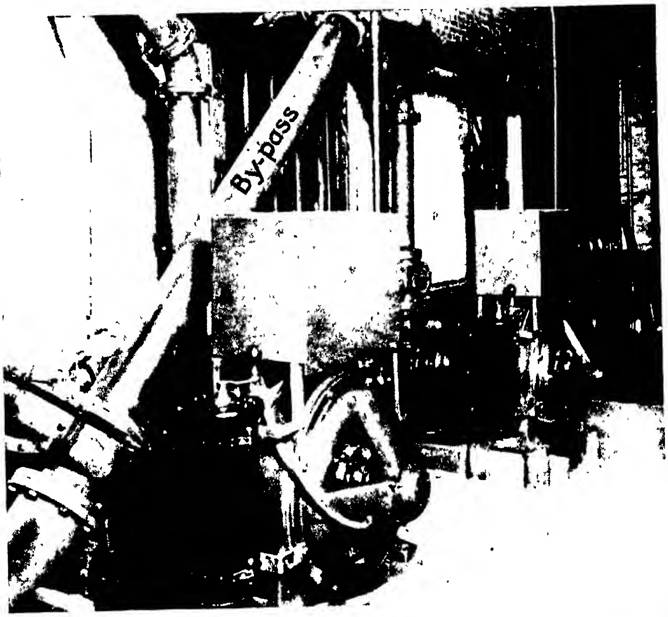


FIG. 119.

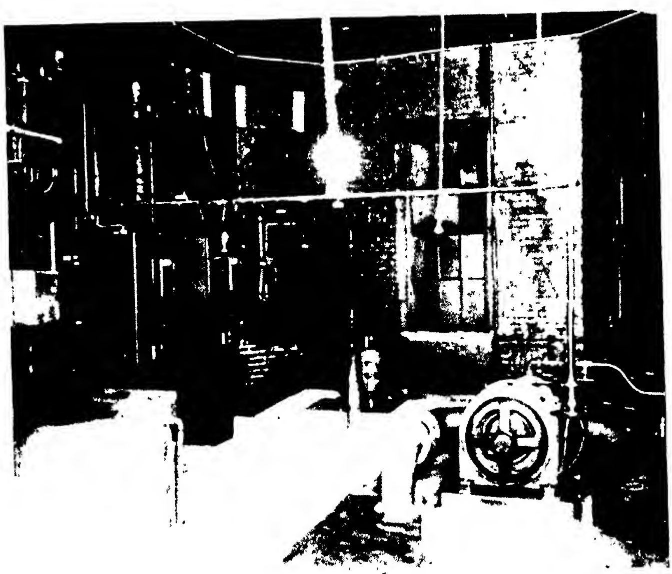


FIG. 120.

the motor control panels. The horizontal pump shown in the foreground is operated as long as the fluid level remains within the suction limit in the pit, and when the inflow does not greatly exceed the capacity of the pump. Conditions other than these require the operation of the other pump, which, as before stated, is of the vertical shaft submerged type and of

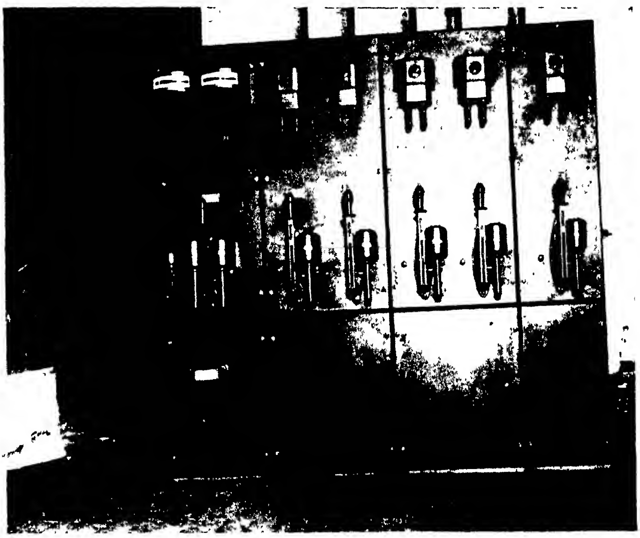


FIG. 121.

larger capacity than the horizontal pump. The smaller unit is operated during the greater part of the time.

Switchboard. — Figure 121 illustrates the switch-board for controlling the operation of the water pump and air compressor motors located in the main building. As shown, it consists of four panels having switches and connections for starting and stopping any one or all motors. A voltmeter and an ammeter are provided so that instantaneous current consumption may be ascertained. An integrating wattmeter is mounted on a sub-panel for registering the total current consumption.

The Wells. — The wells with their surface piping are shown in Fig. 117. The discharge opening of the receiver is connected to a manifold having valves conveniently placed, so that any one or more wells may be pumped, as occasion demands. The well conditions are as follows:

	No. 1	No. 2	No. 3	No. 4
Total depth, ft.	500	500	500	500
Diameter of casings, in.	8	8	8	10
Static head, ft.	31	31	31	31
Pumping head, ft.	68	56	61	66
Capacity, gal. per min.	400	500	600	1200
Operating pressures, lb.	42	42	42	42
Length of air line, ft.	161	149	154	159
Submergence, per cent.	58	67	60	59

It will be noted that the operating pressure of all wells is identical, and since the same is true of the starting pressure, it is

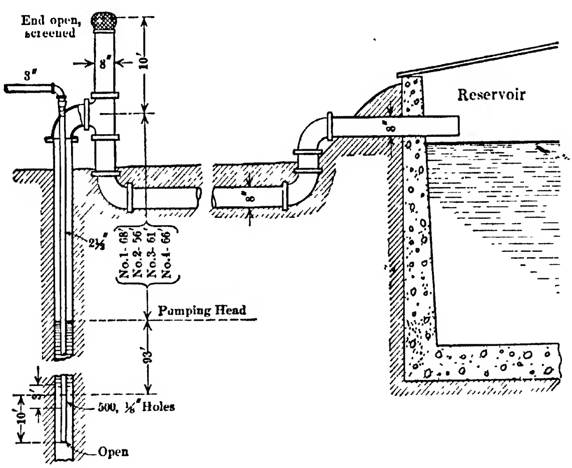


FIG. 122.

possible to start and operate any group of wells without throttling the air to any one. This is a point often overlooked by engineers in designing an air-lift pumping system for a number of wells.

The proper depth of submergence was ascertained by experimenting with varying lengths of air lines and in manner previously described. After determining the most advantageous points of submergence, the wells were balanced by slight changes in air line lengths so as to operate on uniform pressures.

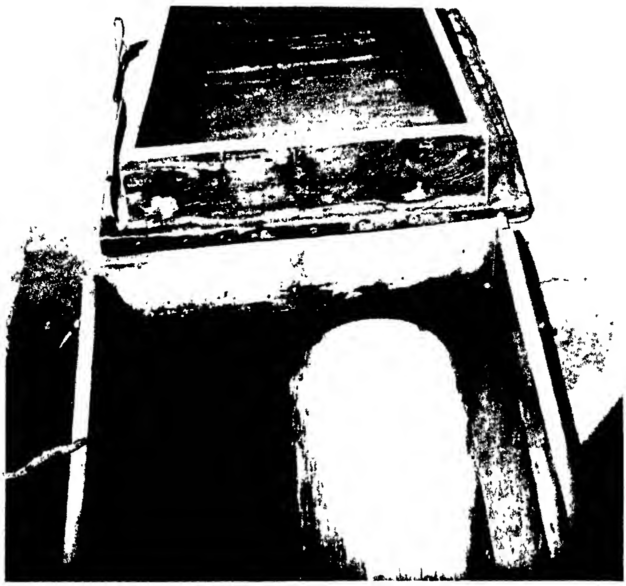


FIG. 123

In Fig. 122 is shown the method of piping the wells. The air and water are separated at the well tops. The air is vented off through the riser pipes and the water flows by gravity to the reservoirs. The object of this is to reduce the velocity of flow and reduce, in proportion, the frictional losses. Another advantage is the elimination of losses incurred by the air " riding " over the water which occurs when an attempt is made to convey the mixture horizontally in a closed piping system.

The lower end of the air lines are open, and ten feet above the

ends a number of $\frac{1}{8}$-in. holes are drilled. The sum of the areas of these holes is greater than the cross-sectional area of the pipes, and since the holes are nearer the surface, practically all the air travels through them, thoroughly aerating the water. The ends are left open so as to dispose of scale or other obstructions and prevent their accumulation and eventual clogging of the small holes.

Figure 123 is a view through an opening in the top of one of the reservoirs of the discharge from the 10-inch well. The flow is exceptionally steady and free from the characteristic pulsations of the poorly designed air-lift.

Operation. — The operating force is required to make and record hourly readings of the electrical instruments and flow meter and submit a daily report on the operation of the plant. From the data contained in a typical report of this kind, the curves shown in Fig. 124 have been plotted.

The *Horsepower* (*Input*) curve includes the energy necessary to operate all water pump and air compressor motors and the dotted addition to this curve represents the energy expended to replenish the storage in the reservoirs. Likewise, the *Water Horsepower* (*Output*) curve represents the total energy realized by lifting the water from the wells with compressed air and forcing it into the mains with the centrifugal pumps. The dotted addition to this curve represents the energy return in replenishing the storage. The *Efficiency* curve is the ratio of output to input and is therefore inclusive of all mechanical, electrical, and air transmission losses. The line loss between the electric generating station and the water-works plant is not included, but this is not justly chargeable to the machinery.

In analyzing these curves, it should be remembered that they are not obtained from test results wherein the load conditions were adjusted to conform to the economical rating of the apparatus; and further, owing to the absence of suitable elevated storage tank, one of the centrifugal pumps at least is called upon to operate continuously and very often under decidedly disadvantageous conditions of load. The *Efficiency* curve represents

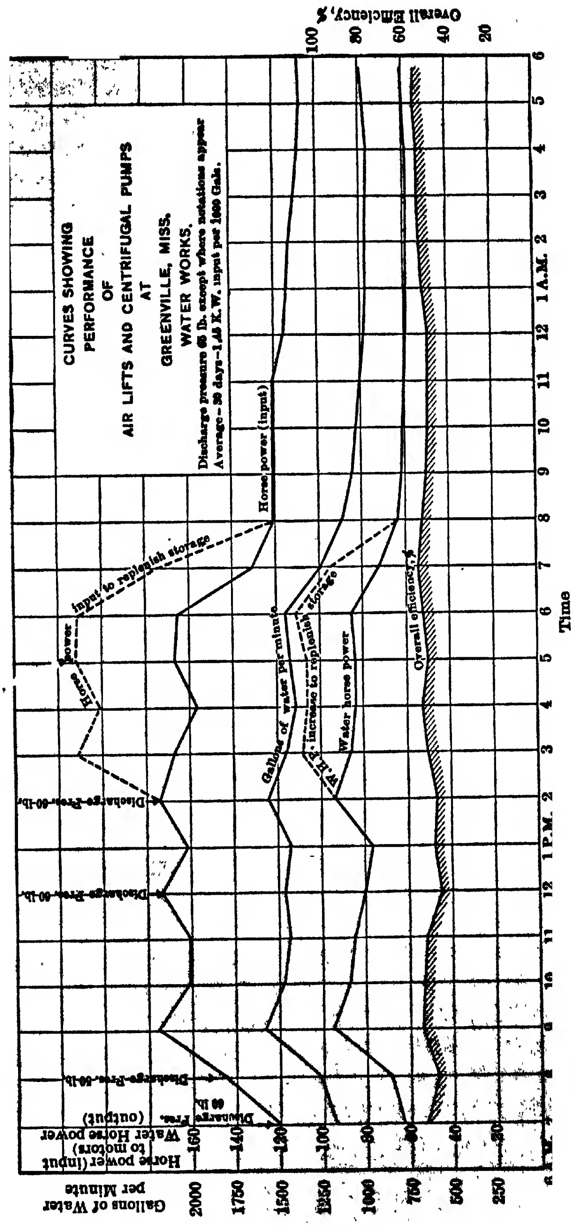

CURVES SHOWING
PERFORMANCE
OF
AIR LIFTS AND CENTRIFUGAL PUMPS
AT
GREENVILLE, MISS.
WATER WORKS.
Overall Efficiency, %
Gallons of Water per Minute
Horse power (input to motors)
Water Horse power (output)
Horse power (input)
Horse power
input to replenish storage
Gallons of water per minute
W. H. P. increase to replenish storage
Water horse power
Overall efficiency, %
Time

FIG. 124.

every-day operating results and such results, more than any other, should be of vital interest to the power plant owner.

The air compressors were manufactured by the Ingersoll-Rand Co., the water pumps by the A. S. Cameron Steam Pump Works, and the sewage pumps by the DeLaval Steam Turbine Co. All electrical apparatus was furnished by the General Electric Co.

INDEX

A

C

P

R

S